Heating · Piping and Air Conditioning

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Steam Heating in the Sky

THE new Douglas DC-4—"world's largest and most luxurious" commercial land plane—is heated by a steam heating system with cabin temperatures governed by a combination of automatic and manual control. General practice has usually been to control temperatures entirely by manual means (this being one of the duties of the stewardess) for vibration has affected control instruments so greatly as to make their use for aircraft service impractical.

The new plane, which is soon to be ready for service tests by the five major air lines for which it was developed, will seat 42 passengers and a crew of five; as a sleeper it will accommodate 30, and includes a luxurious private compartment or "bridal suite," a rest room for the ladies, a dressing room for men, electric kitchen and dining service, hot and cold running water, and even a telephone system capable of connecting to any exchange. The DC-4 is approximately 139 ft from wing tip to wing tip, is nearly 98 ft long, and will have a top speed of 240 mph from four engines that can develop a total of 5600 hp. At cruising speed it can have a range of 2200 miles. It has a newly developed landing gear with a nose wheel designed to permit shorter, smoother landings and safer take-offs. The nose wheel will maintain the plane in flying position while on the ground, which is particularly important with a sleeper plane of this size as the steep inclination otherwise unavoidable is uncomfortable for passengers.

Development of Plane Heating

Early airplanes were heated by a hot air muff around the exhaust pipe, a method which had several faults, chief of which was the danger of carbon monoxide in the heated compartment if the exhaust pipe developed a crack. This has led to the adoption of steam heating for all transport planes and some military ships.

As the windows of a transport plane cannot be opened because of the engine noise that would be admitted, all ventilation must be through air ducts. Air is picked up

The 42 passenger Douglas DC-4 photographed on a test flight

The new Douglas DC-4 was developed for five of the major air lines, is to be ready soon for their service tests. It has a passenger seating capacity of 42, will accommodate 30 as a sleeper plane. Its many features, which make it the "world's largest and most luxurious" commercial land plane, include a telephone system capable of connecting to any exchange . . . . This new giant of the sky is heated by a steam heating system under thermostatic control.

at the nose of the ship and is forced through the duct system by the "ram" of the air. This causes a slight rise in pressure in the cabin which prevents fumes from the engines entering through possible cracks and eliminates danger of CO in the cabin. The pressure also forces the waste air out the exhaust vents in the cabin ceiling, so no fans are needed.

The ducts of the DC-4 are sized to allow approximately 100 cfm per person, this large volume being desirable because at high altitudes the air density is low and a greater volume is necessary to carry off body heat and keep the cabin fresh. This has proved of help in preventing air sickness.

There are really two independent heating systems on the DC-4, with one boiler connected to one radiator and each side functioning independently; this assures against complete heating failures, as each unit can do a fair job by itself. The air is heated on its way to the cabin to a temperature high enough to care for the heat losses and maintain about 70 F in the cabin. Steam radiators or fin coils are used in the ducts and are of such capacity that with an entering air temperature of —40 F a final temperature of 170 F can be obtained. The radiators have about 1 sq ft of face area, heat about 2000 cfm of air, and if flying at sea level can deliver over 400,000 Btu per hr. The two radiators in the DC-4 weigh 72 lb.

Steam is supplied by two flash type water tube boilers, one in each of the two inboard engine exhausts. The exhaust temperatures are about 1300 F, but the evaporation per sq ft of boiler area is high because of the high velocity of the hot gases over the boiler surface. The boilers are designed, of course, to supply the 400,000 Btu per hr required by the radiators. The boilers weigh about 45 lb each.

Each unit operates on seven quarts of water, with only about three quarts actually used and the remaining four quarts kept in reserve in a surge tank to take care of possible leaks. Each quart of water in actual use makes the complete circuit from boiler to radiator and back to the boiler in less than one minute. Owing to the fact that steam temperature decreases with pressure, the system operates at about 24 lb absolute. This pressure is kept regardless of altitude; hence, the gage pressure increases with altitude from about 10 lb at sea level to about 17 lb at the 18,000 ft level.

The quantity of steam is controlled by limiting the amount of water admitted to the boiler. When the pressure in the radiator drops, more water is automatically let into the boiler.

Control of Cabin Temperature

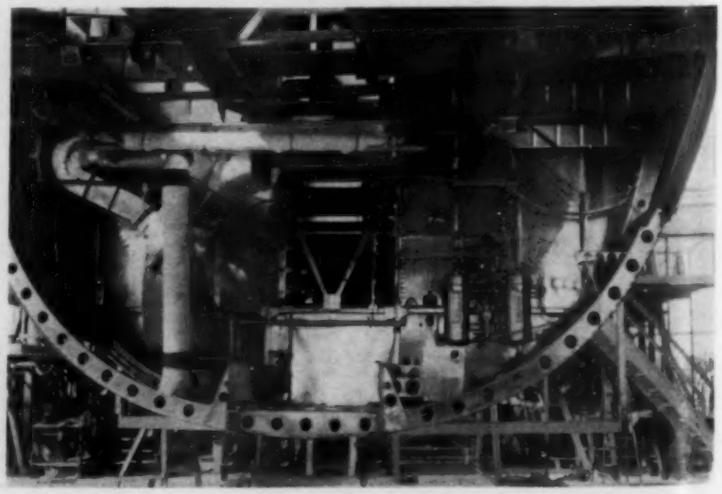
The cabin temperature is controlled by mixing hot air, and cold air bypassed around the radiator, the dampers being governed by thermostat. There is also a manual control which over-rides the automatic in case the automatic is out of order. Bellows filled with thermosensitive liquids are exposed to the outside temperature. These operate the dampers so that there is a definite damper setting for a given outside temperature. Inside the cabin are two electric thermostats which through a mechanism operate with the bellows to make the fine adjustment.

Attemps have been made in the past to control the temperature by the electric thermostats alone, but due to incurable hunting troubles, the device was abandoned.

Complete Air Conditioning Unnecessary

Cooling the air is unnecessary in flight since the ship seldom if ever needs to fly in temperatures above the comfort range except for a very short time after take-off or in landing; while on the ground, portable air conditioners are used. In fact, it is often necessary to use the heating system in summer as well as winter for air temperatures at high altitudes are usually much below those at ground level. When cabins are supercharged for "substratosphere" flying, which is expected as a development for the near future, cooling to some extent by the ventilation air in summer, and the supplying of less heat by the heating system in winter, may be necessary due to adiabatic compression of the entering air—that is, its temperature will be increased by the raising of its pressure from the low atmospheric pressure at high altitudes to the normal atmospheric pressure to be maintained in the cabin.

As to humidification of air in the cabin, no attempts along this line have been made due to the weight of water it would be necessary to carry. Also, studies have indicated that under average flying conditions, cabin



Basement in the sky. Beneath the floor of the DC-4, built by Douglas Aircraft Co. at Santa Monica, Calif., is this maze of controls, hydraulic, heating, electric and ventilating systems for operation of this modern sky giant. The cut-out pictured here fits over the center section of the wing

humidities are generally maintained by natural means close to those assumed to be desirable. Filtering has been found unnecessary because the air through which the ship is flying is nearly free from dust. One air line has found, as a result of some tests, that the pollen count in the plane cabin is below even the low count of the outside air at flying altitudes. This was attributed to the fact that the particles impinge on the duct surfaces as the duct has many turns before the air enters the cabin at individual outlets at the passenger seats.

Vibration a Problem

An important problem in heating planes is that caused by vibration. One air line has developed a flash boiler of corrugated double sheet construction which eliminates the use of tubes, and has been getting good performance from it. Other lines use the tube type boilers, but in this particular case the period of vibration of the engines used by the air line caused failure of boiler tubes no matter how they were constructed. The water used for filling the heating system for each flight causes scale and corrosion trouble, and this same air line is considering installation of water softening and purification equipment at each of its stops.

Another problem is of course that of weight. Commercial parts normally used in heating and ventilating systems are not usually suitable for aircraft use either because of weight or because they are influenced by changes in air pressure due to changes in altitude.

Safety Pays Dividends, Steel Company Finds

THAT safety pays dividends has been proved time and again in the plants of the Jones & Laughlin Steel Corp. since the inauguration in 1926 of a separate department devoted to safety engineering, says H. J. Griffith* in a recent issue of *Industrial Standardization*, which is published by the American Standards Association. Well planned and efficiently managed efforts to bring into effect every means by which accidents can be prevented have resulted in dividends in management's knowledge that employees are safe and comfortable while at work—dividends by increasing efficiency and improving employee relations—dividends by eliminating damage to valuable equipment and materials—dividends in the form of actual cash savings by eliminating accidents that in the past led to expensive lawsuits or payment of compensation because of personal injuries.

The safety factor was given special consideration in the planning and construction of the company's 96 in. continuous mill completed last fall. As a result, the mill was acclaimed by safety engineers during a recent visit as one of the safest they had ever seen.

There are no moving parts exposed where an employee might be injured, unless he violates regulations and deliberately places himself in a hazardous position, according to Mr. Griffith. A unique feature demonstrating the extent to which safety precautions were taken is the roll housings in the hot mill, where the piping to the descaling sprays—which are operated at 1200 lb per sq in. pressure—is shielded to protect employees in case of pipe failure.

Shields over the descaling spray pipes protect workmen in case of failure of the pipes which carry 1200 lb per sq in. pressure

*Manager of Safety and Welfare, Jones & Laughlin Steel Corp.

The world's first air conditioning installation for tea tasters has been made in the sampling and tasting rooms of the London Cooperative Wholesale Society, importers and exporters of fine teas. Tests showed that the sensitivity of the tasters' palate is sharpened by putting them in a cool, clear atmosphere.

Use the New American Pipe Standard

By H. H. Morgan and Sabin Crocker*

UNTIL comparatively recently commercial requirements for wrought iron and wrought steel pipe were fairly well met by the three schedules known as standard weight, extra strong, and double extra strong. The outside diameters, threads, and to some extent the wall thicknesses followed the dimensions laid down in 1886 for "Standard Pipe and Pipe Threads" by an A.S.M.E. committee. For many years this pipe was joined either directly with threaded ends and threaded couplings or fittings, or with screwed companion flanges matching one another or the flanges on fittings. The fact that practically all pipe was threaded in the early days called for heavier wall thicknesses for given service conditions than are required for present day lapped or welded joints where no metal is removed in cutting threads. Furthermore "standard weight" pipe had the thinnest walls practical for use with American Standard (or Briggs) taper pipe threads, no matter how low the working pressure might be. That the old standards were sufficiently serviceable for the limited variety of conditions which they were called on to meet is attested by the few changes registered for many years following 1886.

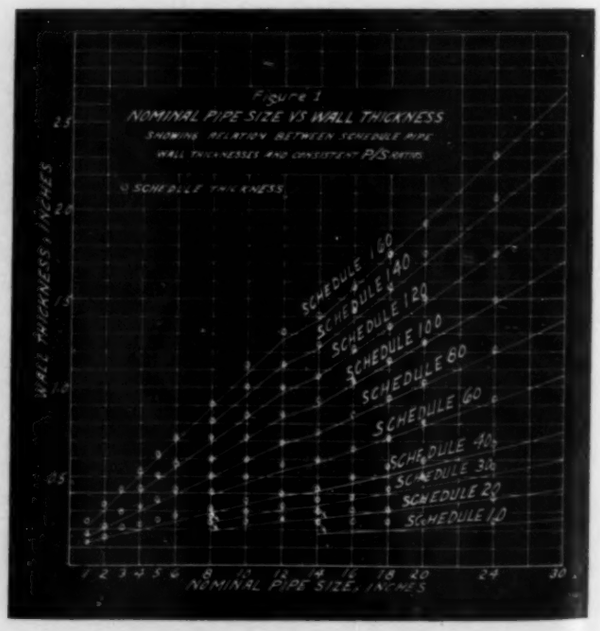
Since about 1920, however, when steam pressures and temperatures started to climb above the 250 lb per sq in. (psi) gage and 600 F which had been tops for the preceding two decades, and when the diversifications of new process work such as the oil and gas industries demanded a greater variety of wall thicknesses and diameters, there has been a tendency to order special thicknesses not included in the three weights comprising the 1886 standards, and sizes larger than listed therein. The problem in manufacturing and stocking an excessive number of wall thicknesses was further complicated by the modern tendency of specifying a wide variety of medium carbon and alloy steels in addition to the original product as made of wrought iron or mild steel, and of calling for new methods of manufacture such as seamless, or resistance or fusion welded as well as the older furnace welded pipe of the lap or butt varieties. With all these possible variations in mind, the advantages became apparent of restricting the choice of wall thickness to a reasonable number of standard schedules which would adequately cover the field in a consistent manner without trying to provide hair-line differences in thickness.

Hence in March, 1927, the American Standards Association, acting on the recommendation of a survey committee, authorized the organization of Sectional Committee B36 on Standardization of Dimensions and Material of Wrought Iron and Wrought Steel Pipe and Tubing. The American Society of Mechanical Engineers and the American Society for Testing Materials were designated as joint sponsors. It is the purpose of this article to review the standard formulated by that committee, known as A.S.A. B36.10 Wrought Iron and Wrought Steel Pipe, and point out its advantages with a view to furthering its use by the piping trade. A.S.A. B36.10 has been an American Tentative Standard since 1935 and is now up for advancement to full American Standard. The Sectional Committee, representing 27 associations, technical societies, and government departments, has approved the new edition of this standard, and it is now being considered by the sponsor organizations, the American Society of Mechanical Engineers and the American Society for Testing Materials. These two organizations will submit their recommendations to the American Standards Association which will then take final action on its approval as American Standard. If the trade is to derive the maximum benefit which might be attained through having the new standard come into general use as soon as possible, it is important that its advantages be widely publicized now.

Table 2 of A.S.A. B36.10 giving the dimensions of welded and seamless steel pipe is reproduced here as Table 1 of this article for illustrating points under discussion. The standard contains a similar table for Schedules 10 to 80 inclusive of wrought iron pipe, and corresponding tables of weights for both products.

Limitations of Old Schedules

The limitations of the old standard weight, extra strong, and double extra strong schedules were fourfold. (1) There was no provision for thin walled pipe. (2) There were no intervening standard thicknesses between the three old schedules which covered too great a spread to be economical without intermediate weights. (3) There were no standardized dimensions for pipe in sizes larger than 12 in. diameter in standard weight and extra strong, nor 8 in. in double extra strong. (4)



*Chairman and Secretary, respectively, of A.S.A. Sectional Committee B36. Mr. Crocker is a Member of H. P. & A. C.'s Board of Consulting and Contributing Editors.

Since none of the old schedules would consistently fit any given set of service conditions throughout very much of its size range, it usually was necessary in choosing pipe of different diameters for any given project to skip around between the three established schedules and even to order special thicknesses which the mills soon came to manufacture for the accommodation of their customers. Each of these shortcomings of the old schedules is of sufficient consequence to warrant elaboration.

Thin Walled Pipe Needed for Welding

As already mentioned, the weakening effect of cutting threads for joining the ends of pipe led to using heavier walls than are necessary where the ends are attached with van stoned joints, flared compression couplings, welding or equivalent methods which do not encroach on the pipe wall. When made up with plain ends of this sort, high grade standard weight (or Schedule 40) pipe is good for pressures of the order of 400 lb per sq in. or more, although when threaded the Code for Pressure Piping allows it only for steam pressures below 250 lb or water pressures under 125 lb where the temperature is above 220 F. The fact that standard weight pipe having plain ends is adequate for use at such relatively high pressures suggested the possibility of effecting noteworthy economy through providing lighter wall schedules where plain ends are used under less severe conditions.

While there are certain considerations tending to set a minimum pipe wall thickness below which it is not advisable to go for fear of not retaining sufficient metal for mechanical strength, resisting corrosion, or for adequate fusion welding, a well defined need existed for pipe of medium to large diameters in wall thicknesses corresponding to Schedules 10, 20 and 30. The practical limitations on thin walled pipe are manifest in Table 1 where no thicknesses are listed below 0.250 in. except in the 4 in. size and smaller. The selection thus afforded provides a sufficient variety from which to choose the allowable thicknesses computed by design formulas such as those of the A.S.A. Code for Pressure Piping and the A.S.M.E. Boiler Code which provide a a constant addition to wall thickness beyond that thickness dictated by pressure and temperature alone.

The power industry is fast becoming accustomed to ordering power piping according to the schedule number system of the American Standards Association B36.10 standard, finds the method works admirably. With B36.10 up now for consideration of its advancement from a tentative to a full American Standard, it is the aim of the Sectional Committee that developed it—and the purpose of this article—to bring the schedule number system to the attention of the piping trade in general in such manner as to expedite its universal use. It is hoped that those responsible for ordering pipe will initiate the practice of specifying what they want by diameter and schedule number

The constant addition to thickness for plain end steel or wrought iron pipe, which varies from 0.02 to 0.10 in., depends on pipe diameter, service conditions, and the particular formula involved. The Code for Pressure Piping contains a further stipulation that the P value for pressure used in its formula shall not be taken at less than 100 lb per sq in. gage for any conditions or material. These provisions afford a sufficient margin to offset the inroads of corrosion and at the same time insure reasonable resistance against crushing, collapsing under external pressure, accidental or unexpected mechanical loads or crumpling between supports.

Another consideration tending to set a minimum limit on wall thicknesses arises in connection with fusion welded line joints. While ideas vary on this point, depending on differences in welding technique, shop practice, etc., it probably is safe to say that few if any organizations desire to weld pipe thinner than 0.125 in. wall, no matter how small the diameter, and that for sizes 2 in. and greater most welders need material having a thickness of at least 0.150 to 0.250 in. to make a sound and properly bonded joint. Hence the somewhat limited selection of light wall pipe provided in Table 1 probably will suffice for some time, the chief opportunity for saving on material over the old standard weight schedule being in sizes 8 in. and larger where the new schedules afford a marked reduction in thickness.

Intermediate Pressures Call for New Thicknesses

Referring again to Table 1 or Fig. 1, it is evident that the old setup did not afford sufficient intervening thicknesses to be economical for use with intermediate pressures. This aggravated the tendency to order non-standard thicknesses better suited to the particular

Table 1—Dimensions of Welded and Seamless Steel Pipe

Nominal Pipe Size	Outside Diam.	Nominal Wall Thicknesses for Schedule Numbers										
		Sched. 10	Sched. 20	Sched. 30	Sched. 40	Sched. 60	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160
⅛	0.405	.....	.....	.....	**0.068**	.....	**0.095**	.....	.....	.....	.....
¼	0.540	.....	.....	.....	**0.088**	.....	**0.119**	.....	.....	.....	.....
⅜	0.675	.....	.....	.....	**0.091**	.....	**0.126**	.....	.....	.....	.....
½	0.840	.....	.....	.....	**0.109**	.....	**0.147**	.....	.....	.....	0.187
¾	1.050	.....	.....	.....	**0.113**	.....	**0.154**	.....	.....	.....	0.218
1	1.315	.....	.....	.....	**0.133**	.....	**0.179**	.....	.....	.....	0.250
1¼	1.660	.....	.....	.....	**0.140**	.....	**0.191**	.....	.....	.....	0.250
1½	1.900	.....	.....	.....	**0.145**	.....	**0.200**	.....	.....	.....	0.281
2	2.375	.....	.....	.....	**0.154**	.....	**0.218**	.....	.....	.....	0.343
2½	2.875	.....	.....	.....	**0.203**	.....	**0.276**	.....	.....	.....	0.375
3	3.5	.....	.....	.....	**0.216**	.....	**0.300**	.....	.....	.....	0.437
3½	4.0	.....	.....	.....	**0.226**	.....	**0.318**	.....	.....	.....	.....
4	4.5	.....	.....	.....	**0.237**	.....	**0.337**	.....	0.437	.....	0.531
5	5.563	.....	.....	.....	**0.258**	.....	**0.375**	.....	0.500	.....	0.625
6	6.625	.....	.....	.....	**0.280**	.....	**0.432**	.....	0.562	.....	0.718
8	8.625	.....	0.250	**0.277**	**0.322**	0.406	**0.500**	0.593	0.718	0.812	0.906
10	10.75	.....	0.250	**0.307**	**0.365**	**0.500**	0.593	0.718	0.843	1.000	1.125
12	12.75	.....	0.250	**0.330**	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 O.D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 O.D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 O.D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 O.D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 O.D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 O.D.	30.0	0.312	0.500	0.625	.....	.....	.....	.....	.....	.....	.....

All dimensions are given in inches.

Thicknesses shown in bold face type in Schedules 30 and 40 are identical with thicknesses for "standard weight" pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for "extra strong" pipe in former lists.

The Schedule Numbers indicate approximate values of the expression $1000 \times P/S$.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions. For tolerances on wall thicknesses, see appropriate material specification.

The sort of pipe being bought to the schedule number system of B36.10 standard. The photo shows manifolds and branch connections in the 10 in., 865 lb, 910 F main steam header built of Schedule 100 carbon-moly pipe and intended to serve as a cross-over between three 75,000 kw turbines (the first of which is now being installed) in an extension of the Delray Plant of The Detroit Edison Co.

service conditions of the job. At one time several years ago the power industry and the pipe mills agreed on several schedules of pipe thickness particularly applicable to power plants, which schedules were designated the 250, 400, 600, 900 and 1350 lb per sq in. steam pressure schedules and published as an appendix to A.S.A. Standard B16e in 1927 and as a table in the 1926 version of A.S.T.M. Specification A106. These pressure rating schedules matched up with those of the A.S.A. Steel Flange Standards and served a very useful purpose until the B36.10 standard now under discussion made its appearance and supplanted them with more universally applicable schedules.

Intervening thicknesses were badly needed also for cross country gas and oil lines which often are hundreds of miles long and where any extra thickness beyond actual requirements runs up the cost unduly. Although much of this material is ordered for special rolling at the mill to the most economical thickness obtainable, nevertheless there is considerable demand for small lots of appropriate thickness which must be obtained from stock. Under the old system those thicknesses listed as standard were too limited to afford any real selection for cross country lines, while the special thicknesses to which such pipe could be rolled were too numerous to encourage stocking. Hence a real need existed for thickness schedules sufficiently comprehensive to encourage ordering and stocking pipe of standard dimensions.

No Sizes Over Twelve Inch in Old Standards

Another serious deficiency in the 1886 standards was the lack of sizes larger than 12 in. in standard weight and extra strong, or larger than 8 in. in double extra strong. While this limited range of sizes may have afforded large enough pipes for boilers, turbines and other equipment of the vintage of 1886, it was far from meeting present-day requirements where low and intermediate pressure pipes of 24 to 30 in. diameter are not uncommon. Prior to the advent of the new standard it had become more or less customary to use a ⅜ in. wall for low pressure pipes in the O.D. sizes of 14 to 30 in. diameter, with somewhat thicker walls stepped up with increase in diameter for pressures above 200 lb or so. The new schedule system provides a much wider choice of wall thickness in the O.D. sizes with a better chance for economy in matching service conditions.

Old Standards Not Consistent for Uniform Pressures

The selections of wall thickness throughout the various diameters of the old standards did not line up well with any uniform pressure for which each of them was suited. This discrepancy was particularly manifest throughout the double extra strong schedule, and to a lesser degree in the 8 to 12 in. diameters, inclusive, of standard weight and extra strong. Referring to Table 1, the thicknesses shown in bold face type for sizes 8, 10 and 12 in. of Schedule 30 were obtained by resurrecting former light weight alternatives of standard weight pipe which had been dropped from that list through Simplified Practice Recommendation No. 57 of the U. S. Department of Commerce as adopted in 1927. There was no thickness of 12 in. standard weight pipe suitable for incorporation in Schedule 40, although thicknesses close enough to be acceptable were found for the smaller diameters of that schedule. The old extra strong list showed a common wall thickness of ½ in. for sizes 7 to 12 in. inclusive which, of course, could not be made to fit any uniform system of progression. It was possible, however, to salvage two of these ½ in. thicknesses by assigning the 8 in. to Schedule 80 and the 10 in. to Schedule 60.

Standardization Will Eliminate Useless Duplication

Due to lack of sufficient variety and size range in the three old standards, the trade came to demand intervening thicknesses and larger diameters in greater abundance than it was economical to manufacture and stock for general use. Naturally each customer wished to buy wall thicknesses best suiting his particular service conditions irrespective of whether there was sufficient other demand for the article to justify it commercially. While with exceptionally large orders, such as for cross country gas and oil lines, it is economical to figure thicknesses closely and order special rollings from the mill, the average customer could not brook the delay involved or pay for resetting the mill to run through a small order.

While the old system encouraged the manufacturers to over-tool their plants in order to fill occasional large orders for nonstandard pipe thicknesses, the very multiplicity of these thicknesses rendered it impracticable to stock them for general sale. Hence, despite the great variety of thicknesses for whose manufacture mill equipment existed, small consumers of pipe often were unable to secure those thicknesses which would have been particularly economical for them to use.

Elimination of unnecessary nominal sizes and a few surplus weights already had been accomplished through the adoption by the trade in 1927 of the U.S. Depart-

ment of Commerce Simplified Practice Recommendation No. 57. Acting under this recommendation and in conjunction with the A.S.A. Sectional Committee on Pipe Flanges and Fittings, the little used nominal sizes of 4½, 7, 9 and 11 in. diameter were discontinued completely, and in addition the 3½ in. diameter in pipe walls heavier than extra strong was dropped. This same simplified practice effected the elimination of commercial thickness lighter than "full weight" in the 8, 10 and 12 in. sizes of the standard weight list where a choice of two or three slightly different thicknesses in the same nominal size formerly existed. While adoption of Simplified Practice No. 57 effected a noteworthy reduction of little needed diameters and weights, its scope was confined to restrictive measures as distinguished from creative planning. Thus the broader field of working out a comprehensive scheme of pipe thickness schedules remained open and was delegated to A.S.A. Sectional Committee B36.

One of the chief advantages of the new B36.10 schedule system lies in the fact that it offers a sufficient variety of thicknesses in a restricted choice of diameters to encourage users to specify standard thicknesses rather than special. This in turn insures the manufacturer sufficient volume of business in these thicknesses to warrant keeping them in jobber's stocks and hence available to the trade without delay or the extra expense of running a special order through the mill. The benefit derived by all concerned is obvious.

Basis of B36.10 Standard Dimensions

The B36.10 standard for wrought iron and wrought steel pipe consists of ten weight and thickness schedules for these materials, of which the lightest wall is Schedule 10 and the heaviest Schedule 160, although Schedule 80 is the heaviest wall so far contemplated for wrought iron. Tabular wall thicknesses were worked out by a basic formula intended for this purpose only and not to be confused with design formulas such as found in the A.S.M.E. Boiler Code and the A.S.A. Code for Pressure Piping. The nature of the basic formula and the method of using it are described in the preface to the B36.10—1935 Tentative Standard and at greater length in an article published in HEATING, PIPING AND AIR CONDITIONING for October, 1933.[1] Tabular values of weights for corresponding sizes of steel and wrought iron pipe are identical in sizes up to and including 12 in., although the nominal wall thicknesses for the two materials have been adjusted slightly to compensate for the small difference in specific weights. If the weights per foot of pipe of the two materials are given as the same, the wall thickness of wrought iron pipe must necessarily exceed those for steel by about two per cent. In sizes 14 in. O. D. and larger where the thicknesses are given the same, the necessary adjustment has been made in the weights.

Significance of P/S Values

Schedule numbers represent approximate values of the expression 1000 P/S, where P is pressure in lb per sq in. and S is the maximum bursting or hoop stress which the material is capable of sustaining with a reasonable factor of safety. For example, Schedule 40 seamless pipe of mild steel having an allowable working (bursting) stress of 10,000 lb per sq in. at around 700 F should be good for just about 400 lb steam pressure at that temperature, as demonstrated by:

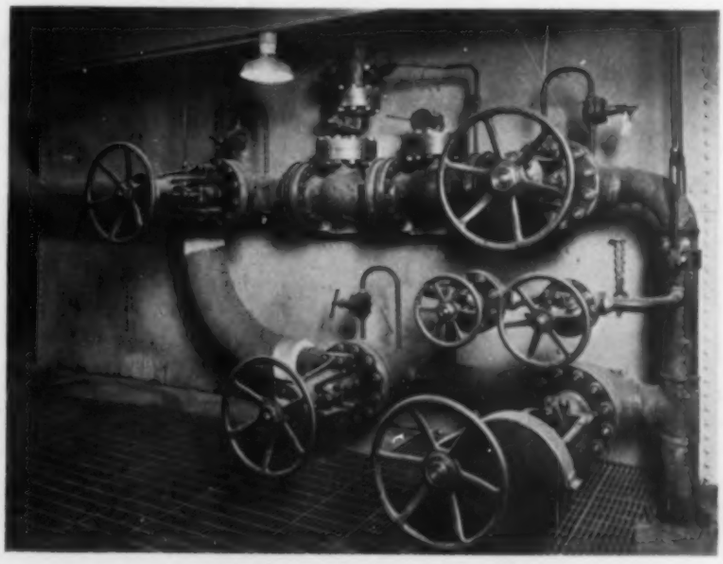
$$\text{Schedule Number} = (1000 \times 400) \div 10{,}000 = 40$$

In the same Schedule 40 wall thicknesses other materials and steam temperatures would, of course, have different allowable bursting stresses and hence be good for different working pressures. The foregoing relation does, however, give an approximate rule-of-thumb for visualizing the physical significance of the schedule numbers in easily understood terms. In a very rough way the allowable working pressure often comes out about ten times the schedule number except where creep is a predominant factor. This is because of the fact that at temperatures up to 750 F. the allowable stress for carbon steels averages about 10,000 lb per sq in. On the other hand, provision for new materials and more severe service conditions than contemplated, even at present, are provided for in that the thicknesses are not tied to particular pressure-temperature conditions or materials as was the case with the old series of thicknesses first published in connection with A.S.T.M. Specification A106-26T.

Presentation of the basic formula itself is purposely omitted here since it already has served its purpose and has been recorded in the aforesaid references where it is available for the information of those interested in the historical side of this project. Further publication of the formula would merely promote confusion through seeming to condone its use for design purposes, whereas the exact contrary is desired as pointed out below.

The basic formula was intended merely for use in computing a rational set of thickness schedules, each of which would come somewhere near fitting a given set of service conditions throughout the full range of pipe diameters embraced in the schedule. After theoretical thicknesses were computed by formula, corresponding

Six inch feedwater regulator and bypass for one of the new boilers of the Delray extension. The boiler feed pipe is Schedule 80 silicon killed carbon steel



[1] "Piping Standardization," by Sabin Crocker, pp. 500-504.

tabular thicknesses were selected from available commercial thicknesses, taking in each case the commercial thickness next greater than the theoretical.

Computation of Wall Thickness for Given Service Conditions

Designers wishing to compute the proper thickness of pipe for a given service application should use the design formulas provided in the A.S.A. Code for Pressure Piping or the A.S.M.E. Boiler Construction Code, as the case may be. These formulas give the minimum pipe wall thickness required on inspection. It is trade practice, however, to furnish pipe to nominal or average wall thicknesses with a plus and minus tolerance to look after variations in manufacture. In ordering pipe, therefore, a nominal wall should be selected from the B36.10 tables which, after deducting for mill tolerance as defined in the specification under which the pipe is bought, is just sufficient to afford the minimum thickness on inspection as computed by the design formula. Procurement of pipe will be facilitated through buying under a standard specification covering properties of the material and including manufacturing tolerances.

Standard Weight and Extra Strong Now Schedules 40 and 80

Fortunately, the old standard weight and extra strong thicknesses coincided rather closely with the new Schedules 40 and 80 respectively as shown in Table 1 where thicknesses common to both systems are set in bold face type. This constitutes a definite advantage, since the use of these old weights is so firmly intrenched in the plumbing and heating trade that any plan which did not embrace them at least through a transition period, would not be generally acceptable. The fact that a few old thicknesses are split between adjoining columns arises from the lack of consistent progression of thickness with increasing pipe size under the old system, as well as to the fact that previous to 1927 there were alternate thicknesses of standard weight pipe in the 8, 10 and 12 in. sizes. Besides being thicker than Schedule 160 pipe, which is the heaviest now contemplated in the standard, the old double extra strong schedule was so erratic as to have no place in the new system and accordingly its thicknesses were dropped from consideration.

Examination of Table 1 shows that whereas all schedules cover the large sizes up to 24 or 30 in. diameter, only Schedules 40, 80, 120 and 160 extend below the 8 in. size, and of these only 40, 80 and 160 have sizes below 4 in. That there is good reason for this is apparent from Fig. 1 where the schedules are portrayed graphically in fan-shaped formation diverging from an origin of small diameter and light wall thickness. The close proximity of the lines representing Schedules 40, 80, 120 and 160 from near the origin out to sizes of the order of 4 to 8 in. diameter demonstrates the reason for omitting the intervening schedules within this range as unnecessary because of the very small differences in thickness involved.

Pipe conforming to B36.10 standard dimensions can be bought to a wide variety of specifications as listed for information in that standard. Among the products contemplated to be furnished to these dimensions are seamless pipe, furnace welded pipe of the lap and butt varieties, resistance welded, forge welded, fusion welded, spiral riveted or welded, and lock bar. Different grades of material are contemplated as well as different finishes such as hot or cold drawn, black or galvanized or other protective coating. These specifications are sponsored by various bodies, such as the American Society for Testing Materials and the American Petroleum Institute. Specifications listed in the reference table of B36.10 have been reviewed by Sectional Committee B36 and accepted as suitable for use in purchasing the products which they cover. Those A.S.T.M. specifications which have the status of a full standard have been adopted also as American Standards and given A.S.A. serial numbers.

Getting the New Standard Into Use

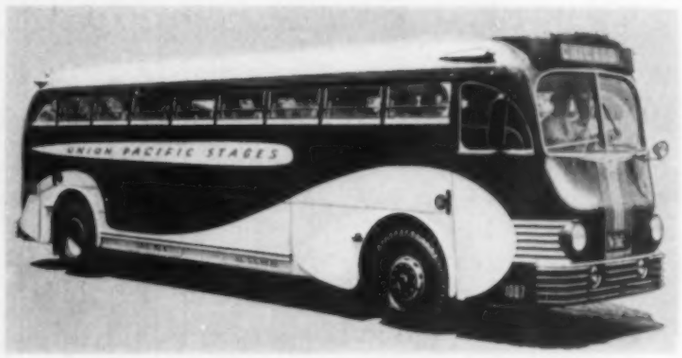
A.S.A. B36.10 was adopted and issued as a tentative American Standard in 1935 and recommended by Sectional Committee B36 in 1938 for advancement to full American Standard, substantially unchanged. The old system of standard weight, extra strong and double extra strong pipe is so firmly intrenched with the trade, however, that thus far it has seemed impracticable to set a definite date for a complete change-over to the new schedule numbers. In the meantime the advantages of the new standard are being demonstrated to the trade with a view to gradually shifting the bulk of purchases from the old system to the new. The fact that most thicknesses of standard weight and extra strong pipe have been carried on in Schedules 40 and 80 as shown in Table 1 helps to make feasible this gradual change-over.

As a first step, Table 1 was incorporated in A.S.T.M. Specification A106-36 for Lap Welded and Seamless Steel Pipe for High Temperature Service, which is largely used for high pressure power plant work at temperatures up to 850 F. Also certain sizes and thicknesses taken from Table 1 were selected by the pipe manufacturers and the Prime Movers Committee of the Edison Electric Institute as standard for stock purposes in carbon-molybdenum pipe and listed in an appendix to A.S.T.M. Specification A206-37T for that product. The power industry is fast becoming accustomed to ordering power piping according to schedule number and finds the new system works admirably.

Another step in encouraging use of the new standard will be publication of Table 1 and a corresponding table of weights as an appendix to A.S.T.M. Specifications A53 and A120 for the more commonly used grades of steel pipe. These specifications already contain the old tables for standard weight, etc., pipe in the body of the specification where they will be retained for the time being. In another year or so, however, it is hoped that the pipe mill men and the consumers of this product will be agreeable to reversing the arrangement of these tables in A53 and A120 so as to put the new schedules in the body of the specifications and relegate the old standard weight, etc., tables to the appendix, eventually dropping them altogether as the schedule system takes hold.

It is the aim of Sectional Committee B36 and the purpose of this article to bring the new system of schedule numbers to the attention of the piping trade in general in such a manner as to expedite universal use of the B36.10 standard through familiarizing all concerned with its advantages over the old set-up. With this end in view it is hoped that those responsible for ordering pipe will initiate the practice of specifying what they want by diameter and schedule number so as to effect the gradual change-over desired.

Bus Air Conditioning



Once the railroads began to adopt air conditioning, development was rapid until today it is practically standard equipment; certainly no new streamliner would be built without it. . . . As did the railroads a few years ago, the bus industry this year began to offer comfort air conditioning to its patrons. If air conditioned bus travel attracts the public as does air conditioned rail travel, next year should see hundreds of buses featuring this new service in operation. . . . Buses present special requirements which affect the air conditioning design, and these are illustrated in this analysis by Jerry Hicke*. The author has been closely concerned with the subject for many months and has travelled extensively testing and demonstrating one of the air conditioned buses

IT is generally true that the most comfortable conditions for the average person consist of a dry bulb temperature of from 76 to 80 F with a relative humidity of 45 per cent. These conditions are maintained in air conditioned buses just as they are in air conditioned railroad cars or buildings. Outside air must be provided for ventilation, and, due to the fact that in the average bus there are a large number of people in a small enclosure, 20 to 25 per cent of the total air delivered to the conditioned space comes from outside.

There are many different styles of buses, each of which presents particular problems in so far as load calculations are concerned. However, any one style of bus should illustrate and involve most of the problems encountered in all.

Conduction Heat Gain

The calculation of the conduction heat gain for a bus is very little different from that for a building. The fundamental equation $H = UA\ (t_o - t_i)$ is used[1], but the difference lies in the coefficient, U. In the ordinary building, the film coefficient for the outside air is figured at an air velocity of 15 mph, whereas in the case of the bus the velocity of the air on the outside surfaces is usually three or four times that amount. Hence, the film coefficient for outside air, or the amount of heat passing from the outside air to each square foot of the outside wall surface per hour, is increased. Despite this increase, the total effect on the wall heat gain is very small.

Most buses have single glass windows, and walls consist of steel on the outside, insulation of the felt type, and steel again on the inside. The floor and ceiling are of approximately the same construction as the walls so it is generally assumed that they have the same U value.

For the windows (using the well-known formula):

$$U = \frac{1}{\frac{1}{f_o} + \frac{1}{k_g} + \frac{1}{f_i}}$$

Outside air film: $f_o = 10$; $\frac{1}{f_o} = \frac{1}{10} = 0.1$

Single glass: $k_g = 7.5$; $\frac{1}{k_g} = \frac{1}{7.5} = 0.133$

Inside air film: $f_i = 1.65$; $\frac{1}{f_i} = \frac{1}{1.65} = .61$

$$U = \frac{1}{0.1 + 0.133 + 0.61} = \frac{1}{0.843} = 1.19$$

Thus, we have a glass heat transfer coefficient of 1.19 (say 1.2) instead of 1.1, which is an increase of 8 per cent over that ordinarily used.

Assuming that the walls of the bus are constructed of $\frac{1}{16}$ in. steel, 1 in. hair felt, and $\frac{1}{16}$ in. steel, the wall coefficient becomes:

$$U = \frac{1}{\frac{1}{f_o} + \frac{x_s}{k_s} + \frac{x_f}{k_f} + \frac{x_s}{k_s} + \frac{1}{f_i}}$$

Outside air film: $f_o = 10$; $\frac{1}{f_o} = \frac{1}{10} = 0.1$

1/16 in. steel: $k_s = 308$; $\frac{x_s}{k_s} = \frac{0.0625}{308} = 0.0002$

1 in. hair felt: $k_f = 0.24$; $\frac{x_f}{k_f} = \frac{1}{0.24} = 4.17$

Inside air film: $f_i = 1.65$; $\frac{1}{f_i} = \frac{1}{1.65} = 0.61$

$$U = \frac{1}{0.1 + 0.0002 + 4.17 + 0.0002 + 0.61}$$

$$= \frac{1}{4.8804} = 0.205 \text{ (say 0.21)}.$$

For the conduction heat gain the temperature difference $(t_o - t_i)$ depends upon design conditions. However, in figuring the floor load, the temperature difference is increased 10 deg to allow for equipment load and radiation from the hot pavement.

The calculation of the sun load for a bus differs only slightly from the calculation for a building. The sun load is figured for one-half of the wall, and the entire roof. In addition, the sun load is figured for the glass in one wall, in the side door, and in the windshield. Maximum temperature differences are always figured in the sun load calculation.

*Transportation Air Conditioning Division, The Trane Co.

[1] H = Btu per hr; U = Btu per hr per sq ft per F; A = sq ft of wall area; t_o = outside temperature; t_i = inside temperature.

Fig. 1—Cooling load estimate sheet

JOB NAME 36 Passenger Bus LOCATION

ROOM WIDTH LENGTH HEIGHT FLOOR

OUTSIDE DESIGN CONDITIONS: DRY BULB 95° WET BULB 76° DEW POINT 68.5° R. H. 41%

INSIDE DESIGN CONDITIONS: DRY BULB 80° WET BULB 65.3° DEW POINT 57.3° R. H. 45%

TIME A. M. 2 P. M. LAT. 40° WALL COLOR: DARK ■ LIGHT □ MEDIUM □

WINDOWS: SHADES ■ AWNINGS □ GAIN □ ROOF COLOR: DARK ■ LIGHT □ MEDIUM □

ITEM NO.	ITEM	DIMENSIONS	AREA SQ. FT.	U	D	SENSIBLE HEAT B.T.U. PER HOUR
	CONDUCTION HEAT GAINS					
1	EXTERIOR WALL, GROSS		444	[illegible]	••	••••••
2	EXTERIOR GLASS		90	1.2	15	1620
3	EXTERIOR WALL, NET		354	.21	15	1115
4	PARTITIONS, NET					
5	GLASS IN PARTITIONS					
6	FLOOR		220	.21	25	1155
7	CEILING OR ROOF		220	.21	15	693
8	MISC.					
9	TOTAL CONDUCTION HEAT GAINS	•••••	•••••	••	••	4583
	EXCESS SOLAR HEAT GAINS					
	WALLS FACING:					
10	NORTHEAST					
11	EAST					
12	SOUTHEAST					
13	SOUTH					
14	SOUTHWEST		177	.21	11	409
15	WEST					
16	NORTHWEST					
17	ROOF		220	.21	65	3003
	GLASS FACING:					
18	NORTHEAST					
19	EAST					
20	SOUTHEAST					
21	SOUTH					
22	SOUTHWEST		55	1.2	60	3960
23	WEST					
24	NORTHWEST					
25	SKYLIGHTS					
26	MISC.					
27	TOTAL EXCESS SOLAR HEAT GAINS	•••••	•••••	••	••	7372
	DUCT HEAT GAINS					
28	INSULATED DUCTS					
29	NON INSULATED DUCTS					
30	TOTAL DUCT HEAT GAINS	•••••	•••••	••	••	

ITEM NO.			HEAT LOAD B.T.U. PER HOUR SENSIBLE	LATENT
	BODY HEAT GAINS			
31	SENSIBLE NO. OF PEOPLE 37 X 225		8325	
32	LATENT (QUIET) NO. OF PEOPLE 37 X 175			6475
33	LATENT (ACTIVE) NO. OF PEOPLE X 250			
34	TOTAL BODY HEAT GAINS			
	EQUIPMENT HEAT GAINS			
35	ELECTRIC LIGHTS. WATTS X 3.4			
36	SMALL ELECTRIC MOTORS. TOTAL H.P. 166 W. X 3.4		564	
37	LARGE ELECTRIC MOTORS. TOTAL H.P. X			
	ELECTRIC EQUIPMENT			
38	NO. X			
39	NO. X			
40	NO. X			
41	NO. X			
	GAS EQUIPMENT			
42	NO. X			
43	NO. X			
44	NO. X			
45	NO. X			
	WARM PIPES			
46	FT. X			
47	FT. X			
48	TOTAL EQUIPMENT HEAT GAINS			
	INFILTRATION HEAT GAINS SENSIBLE FACTOR	LATENT FACTOR	•••••	•••••
49	ROOM VOLUME	CUBIC FEET		
	MISCELLANEOUS HEAT GAINS			
50				
51				
52	TOTAL MISCELLANEOUS HEAT GAINS			
	SUMMARY OF HEAT GAINS			
	ITEM NO.	ITEM		
	9	CONDUCTION LOAD	4583	
	27	SUN LOAD	7372	
	30	DUCT LOAD		
	34	PEOPLE LOAD	8325	6475
	48	EQUIPMENT LOAD	564	
	49	INFILTRATION		
	52	MISCELLANEOUS		
53	TOTAL SENSIBLE HEAT GAINS		20844	••••
54	TOTAL LATENT HEAT GAINS		6475	← 6475
55	TOTAL HEAT GAINS = ITEM 53 + ITEM 54		27319	•••••
56	SENSIBLE HEAT PERCENTAGE = ITEM 53 / ITEM 55 X 100 = 20844 / 27319 X 100 =		76.3 %	
57	DRY BULB TEMPERATURE OF AIR SUPPLY = 52.4 DEGREES			
58	WET BULB TEMPERATURE OF AIR SUPPLY = 51.8 DEGREES			
59	RISE IN DRY BULB TEMPERATURE OF AIR SUPPLY = ROOM D.B. = ITEM 57 = 80 — 52.4 = 27.6 DEGREES			
60	TOTAL AIR SUPPLY = ITEM 53 / 1.1 X ITEM 59 = 20844 / 1.1 X 27.6 = 687 or approx. 700 C. F. M.			
	HEAT LOAD OF VENTILATION AIR			
	NO. OF PEOPLE X C.F.M. PER PERSON =	C.F.M. OF OUTSIDE AIR		
61	C.F.M. OUTSIDE AIR 175 X 44.1 B.T.U. PER C.F.M. =	7717 B.T.U. PER HOUR		
	TOTAL COOLING LOAD ON COILS AND REFRIGERATING APPARATUS			
		ITEM 55	27319	
		+ ITEM 61	7717	
62	TOTAL COOLING LOAD, B.T.U. PER HOUR		35036	
63	TONNAGE EQUIVALENT OF COOLING LOAD = ITEM 62 / 12,000 = 35036 / 12,000 =		2.92 TONS	

UNITS SELECTED

A Typical Calculation

For illustration of load calculations, assume a typical bus on which air conditioning has been installed. It has a net wall area of 354 sq ft, glass area of 90 sq ft, roof area of 220 sq ft, and a floor area of 220 sq ft. Assuming this bus to have a route through Missouri, Kansas, and Oklahoma, the average outside design conditions will be 95 F dry bulb temperature and 76 F wet bulb temperature. With inside design conditions of 80 F dry bulb and 45 per cent relative humidity, the temperature difference for all surfaces for conduction heat gain will be 15 F with the exception of that for the floor, which will be 25 F. Using 0.21 as the value of U for the walls, roof, and floor, and 1.2 as the value of U for the glass, the cooling load estimate sheet in Fig. 1 shows the conduction heat gain to be 4583 Btu per hr.

In calculating the sun load, cooling requirements are raised considerably due to the hot surfaces of the bus exposed to direct sunlight. The amount of the heat gain due to the sunlight varies, of course, depending upon the direction of the bus, time of day, latitude, and color the surfaces are painted.

Two p. m. was selected as the time to compute the solar heat gains because the solar heat gain through the roof is generally a maximum at this particular time. As a rule, though not always, the solar heat gain will be a maximum at approximately this time in any enclosure in which there is a relatively extensive roof area.

Inasmuch as a bus is mobile and can travel in any direction, it is evident that for design purposes the bus must be assumed to be traveling in such a direction that the solar heat gain will be a maximum. For this reason, the bus was assumed to be traveling in such a direction as to present its greatest window area toward the southwest. At 2 p. m. the highest solar heat gain is obtained through the vertical surfaces facing in this direction.

Using the same U coefficients as for the conduction load, the sun load proves to be 7372 Btu per hr, Fig. 1.[2] This is nearly double the conduction heat gain.

The average person when inactive gives off heat at the rate of 400 Btu per hr of which 225 Btu is sensible heat and 175 Btu latent heat. Fig. 1 shows the total body heat gain on this basis.

All of the equipment except the air cooling units is situated outside of the conditioned passenger space. Therefore, the only equipment heat gain is due to the small motors on the fans.

Volume of Air to Circulate

The total sensible heat to be removed per hour thus is 20,844 Btu. The problem, then, becomes one of finding the volume of air to be circulated by the fans and the corresponding delivery temperature necessary from the equation:

$$H_s = 60 \times Q \times d_a \times c_a \ (t_r - t_d)$$

where H_s = sensible heat to remove, Btu per hr;
Q = cfm;
d_a = density of the air;
c_a = specific heat of air;
t_r = room dry bulb temperature;
t_d = dry bulb temperature of delivered air.

The dry and wet bulb temperatures of the air supply given in Items 57 and 58 of Fig. 1, are read at the point on the psychrometric chart of Fig. 2 where the sensible heat percentage line is intersected by the curve showing the change in the condition of the air as it flows through the cooling coil. This curve is determined by test or from performance data supplied by the maker of the coil.

The required final dry and wet bulb temperatures of the air supply as read on the psychrometric chart (Fig. 2) at the point marked *air leaving the coil* is 52.4 deg dry bulb and 51.8 deg wet bulb. Inasmuch as a dry bulb temperature of 80 deg is to be maintained inside the bus, the rise in the dry bulb temperature of the air supply will be 80 — 52.4, or 27.6 deg. Knowing the temperature rise of the air supply, the volume of air required can now be computed by means of the above equation. The sensible heat gain of the room as computed in Fig. 1 is 20,844 Btu. Making these substitutions in the preceding equation, the air quantity required is found as follows:

$$20{,}844 = 60 \times Q \times 0.075 \times 0.24 \ (80 - 52.4)$$

$$Q = \frac{20{,}844}{60 \times 0.075 \times 0.24 \times 27.6} = 700 \text{ cfm.}$$

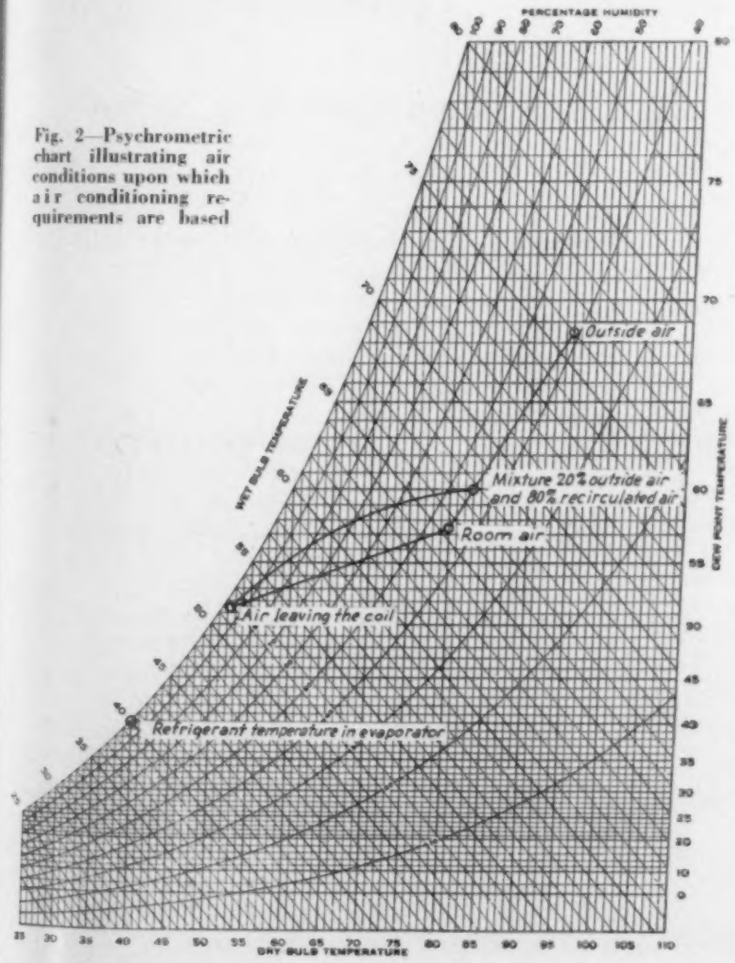


Fig. 2—Psychrometric chart illustrating air conditions upon which air conditioning requirements are based

[2]The values for D of Fig. 1 for computing these solar heat gains were taken from the tables on pp. 276 and 277 of the "Trane Air Conditioning Manual."

Item 60 of Fig. 1 is a condensed form of the preceding equation and computation.

So far nothing has been said about the load due to outside air. Since the total heat of air depends entirely upon the wet bulb temperature, the equation $H_t = 60 \times d_a \times Q \times (h_o - h_i)$ may be used either to calculate the additional load due to outside air or the total load on the cooling equipment. In the former calculation, h_o is the total heat of air at the wet bulb temperature of the outside air, h_i is the total heat of air at the wet bulb temperature of the conditioned space, and Q is the cfm of outside air. In the latter calculation, h_o is the total heat of air at the wet bulb temperature of the air entering the coils, h_i is the total heat of air at the wet bulb temperature of the air leaving the coils, and Q is the total cfm circulated by the fans.

The quantity of outside air to be introduced into the bus is determined by considerations of ventilation. If 25 per cent of the 700 cfm of total air circulated is used, the outside air requirement is 175 cfm.

In Item 61 of Fig. 1, the load due to the ventilation air was figured by using the value of 44.1 Btu per cfm which was computed by assuming that the air is cooled from an initial wet bulb temperature of 76 deg to a room wet bulb temperature of 65.3 degrees.

The total load on the equipment for the bus under consideration amounts to 35,036 Btu per hr or 2.9 tons of refrigeration (Fig. 1).

Method of Air Distribution

The problem of air distribution in bus air conditioning has proved difficult and has required a good deal of experimentation. The air introduced in our example is approximately 9 F lower than normal and consequently makes the air distribution within the bus a considerably more important problem than usually. The fact that most buses have baggage racks above the seats makes it difficult to obtain good distribution from a central duct or ducts along the side walls. In addition, ductwork puts rather high static pressure on the fans. Since it is necessary to use very small units and consequently small fans, the horsepower required to move the air through ducts is high. Not only are the power costs increased but the static pressure due to ductwork makes it even more difficult to obtain good distribution.

Good results have been obtained by discharging the air directly from the fans into a furred down ceiling. This ceiling is perforated, thereby providing even distribution of the chilled air. In addition, the floor load is immediately absorbed and the delivery temperature raised, giving less chance of cold drafts. The recirculation grille is usually placed in the ceiling as close to the unit as possible.

New Hospital to Have Twelve Air Conditioned Operating Suites

WORK is progressing on what has been planned to be one of the finest air conditioned hospitals in the country—the new surgical section of the George Robert White House, Massachusetts General Hospital, Boston. There will be 12 completely air conditioned operating suites. Combined studies of the architects, engineers and hospital authorities will provide the most healthful atmosphere for both patients and surgeons. Coolidge, Shepley, Bulfinch and Abbott, nationally known architects, designed the building. The air conditioning system was planned by the Office of Hollis French, consulting engineers.

One of the features of the air conditioning system will be the double cleaning of the air, which will first pass through automatic, self-cleaning filters, thence through a water spray to be thoroughly washed, and finally through renewable air filters at each air outlet. When patients having contagious diseases are in the operating rooms, additional filters will be placed on the air exhaust grilles. Not only is the removal of air borne impurities assured by this method, but also possibility of bacteria lodging in the air ducts is eliminated.

It was determined that a temperature of 80 F with a relative humidity of 55 per cent was most desirable for operating, this humidity preventing danger of accidental explosions of anaesthetic gases. These conditions will be maintained throughout the year by means of automatic controls. For economy, automatic bypass dampers will be installed to permit recirculation of air while the rooms are being prepared for operations. When any of the operating rooms are in use, however, the entire air supply will be taken from outdoors, with 12 complete changes an hour.

Five spray washer type central station units will be used, the temperature of the water being regulated in summer by means of two "Freon" refrigerating compressors, each of 50 tons cooling capacity. Supplementary heaters in ducts supplied with steam from the Edison company's central steam plant will heat the air in winter, obviating the use of radiators and leaving the walls free for complete cleaning. A separate fan on each unit of the five will draw the air through the sprays and distribute it through ducts to outlets directly over the lighting fixtures in the rooms. Evaporative condensers will be used to conserve water and keep down water costs. The entire equipment, including the compressors, will be under full automatic control.

All pipes are to be equipped with vibration eliminators and compressor, fan and condenser mountings will be on springs. Each of the five central units will have a remote control switch at the nurse's station.

High Pressure Steam Adopted for Heating General Foods' New Mill

GENERAL FOODS CORPORATION'S new corn mill—situated in the heart of the corn country at Kankakee, Ill.—is believed to be the first complete new corn mill built in the United States during the last 20 years. It is a multi-story beam and girder type reinforced concrete structure of modern design which emphasizes the materials and structural features as the basis of the architectural design. The concrete spandrels, with continuous sash, produce a clean-cut and interesting exterior without recourse to ornament in any form. The plant is heated by high pressure steam, with propeller type unit heaters on lower floors and convector type radiation on upper stories.

Structural Features of Plant

One of the most interesting features in the design of this building are the two transverse expansion joints throughout its entire height. This results in the advantages of three separate buildings combined into one, according to the engineers and builders, and is an economical and practical solution to the problem of expansion and contraction in localities where extreme temperature variations occur.

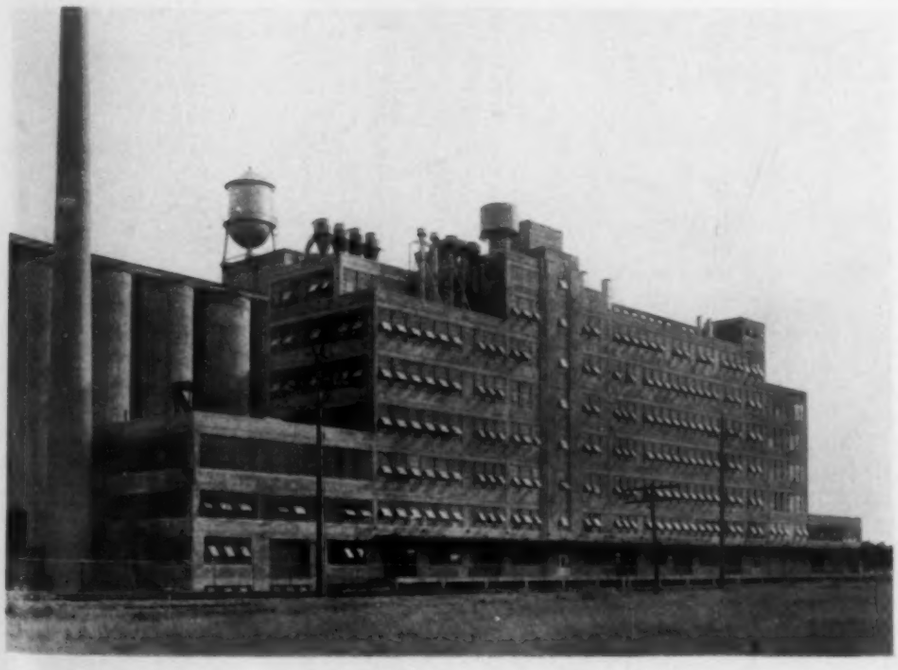
The building is 48 ft wide by 405 ft long, eight stories and basement, with monitor. It is equipped to handle 20,000 bushels of corn a day and is designed so that capacity can be increased to 30,000 bushels. There is a 1,250,000 bushel raw corn storage consisting of 24 concrete silos 24 ft in diameter by 100 ft high; the elevator head house is equipped to deliver the kernel corn from the hopper into the silos, reclaim it from the storage silos and convey it over a bridge to a house bin in the mill, from which point it starts on its way to processing. Fifty-six bucket elevators in the mill feed the various machines, including 48 grinding mills.

Corn (on the cob) is trucked directly to the mill by the farmers, in which case it must first go through the sheller, or is delivered already shelled by railroad car from central grain storages situated throughout the corn belt. About 60 per cent of the corn received is processed and manufactured for the market at the mill, the products being hominy grits, corn meal, corn oil, brewers' flakes, and cattle food as a byproduct. The balance of the mill capacity is sent to General Foods' main plant at Battle Creek, Mich., in the form of coarse grits, the first step in the process of making the breakfast food "Post Toasties." White corn only is used for "Post Toasties," as the processing of yellow corn brings out an objectionable dark brown color.

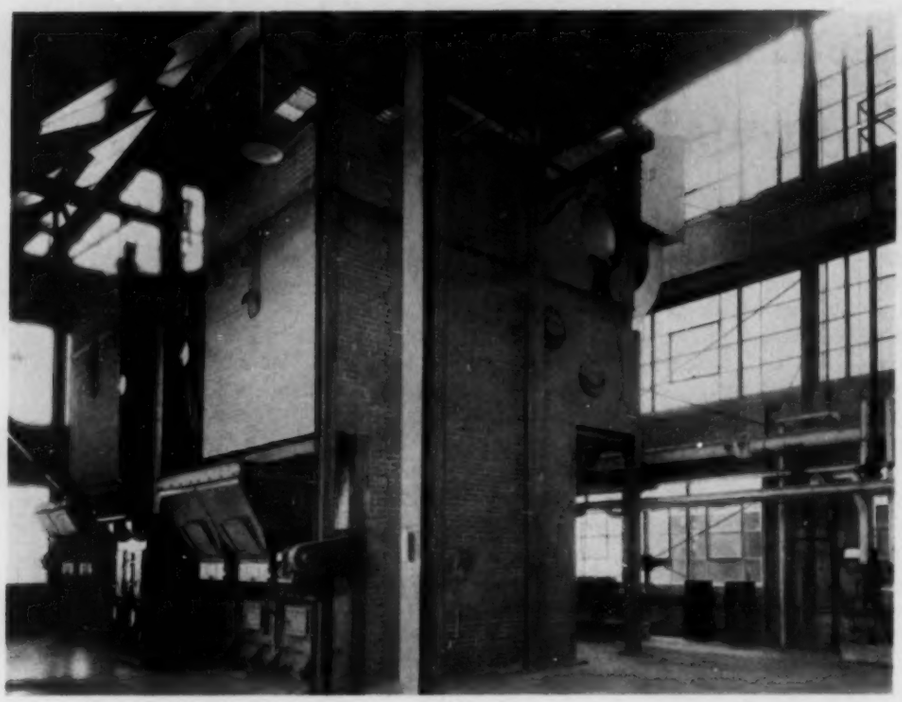
Steam Generated at 150 Lb

Adjoining the mill building is a three story boiler house with two low head, bent tube type, three drum boilers set over overfeed motor driven stokers; space for an additional boiler is provided for use when the mill capacity is increased. This steam generating plant supplies steam at 150 lb pressure for both process and heating use; in winter, the steam requirement is around 12,000 lb per hr, and in summer it is about 7000 or 8000 lb.

Steam for plant heating is supplied through a diaphragm operated, single seat pressure reducing valve



Believed to be the first new corn mill built in the United States during the last 20 years, General Foods Corporation's new mill in the heart of the corn country at Kankakee, Ill., includes interesting structural and mechanical features . . . As steam at high pressure was required for process use, and as steam and electrical requirements are so far out of balance as to make generation of byproduct power not attractive at present, 50 lb steam is used in the radiation and unit heaters . . . A similar scheme is being used at General Foods' new coffee plant at Hoboken, N. J.



The boiler room. Among the instruments on the board between the two boilers is the pressure recorder control instrument governing the remote air operated reducing valve which supplies steam for the heating system

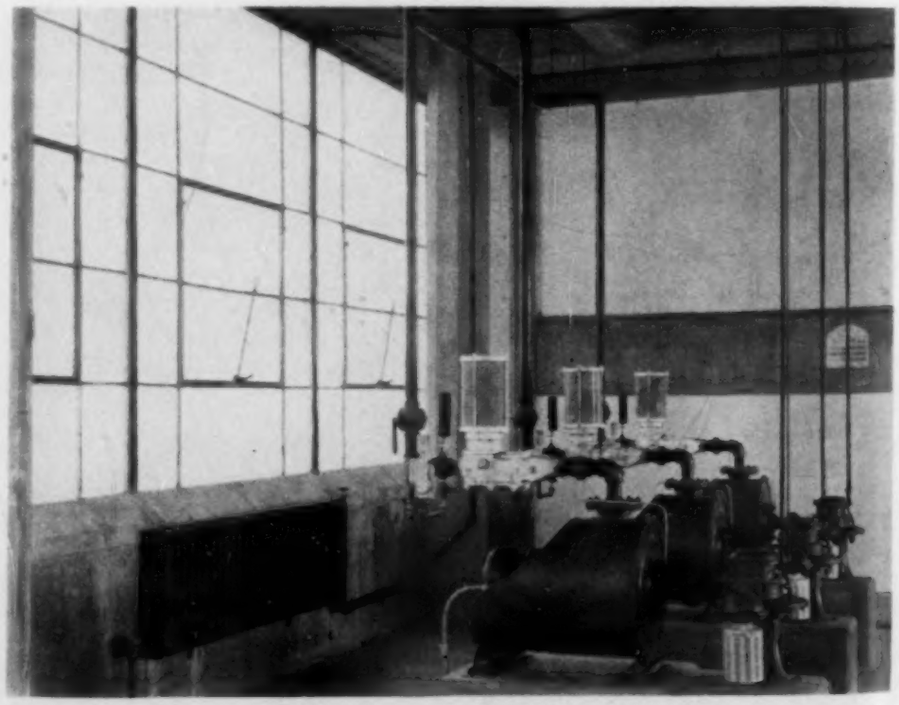
with a rated capacity of 7500 lb per hr which reduces the steam pressure to 50 lb, the pressure carried in the radiation and the unit heaters. The use of high pressure steam for heating was decided upon as a result of a number of circumstances peculiar to the project.

The large amount of glass area in this building made the installation of cast iron wall radiation impractical, as it would have been necessary to have installed two and three tiers of such radiation. Due to the large amount of equipment interference it was not considered desirable to adopt unit heaters for general heating throughout the plant, although there are 24 propeller fan unit heaters in basement, first and second floor areas.

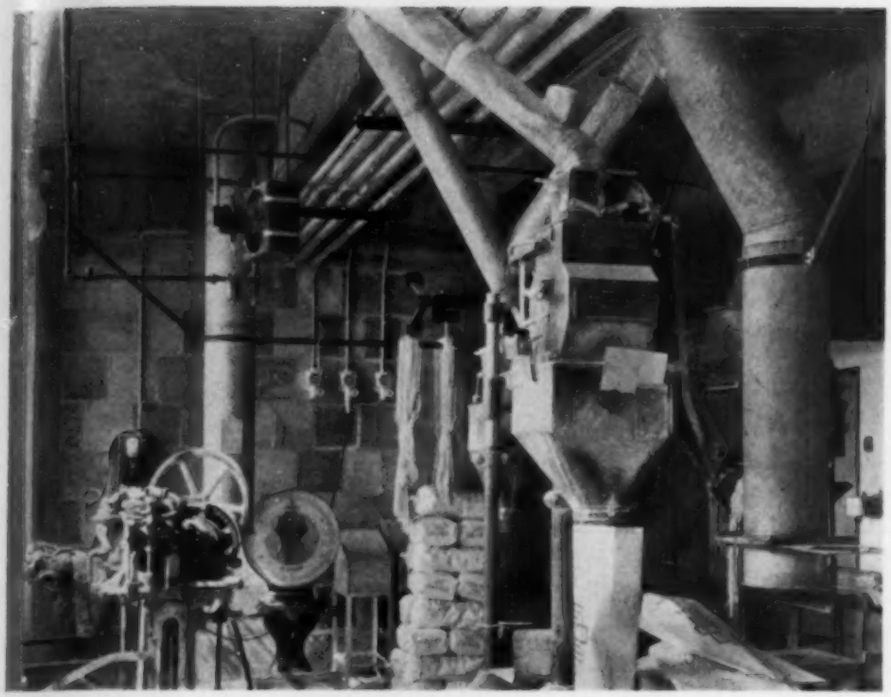
In order to avoid the difficulties mentioned above, it was decided to use high pressure type convector radiators which permitted installing the required heating surface in the minimum space. These radiators are capable of withstanding comparatively high pressures; furthermore, the surface of the radiator subjected to being touched by a person is at a low temperature at high pressures, eliminating the objection of anyone being burned.

It was necessary to install the steam generating plant to furnish steam at 150 lb per sq in. for process use. The electrical and steam loads are so far out of proportion that an installation for the generation of byproduct power is not attractive at present. Having the steam at a high pressure and radiators that would withstand high pressure, it was decided to operate them at relatively high pressures. This afforded the advantage of decreasing the size of the steam mains and the branch piping. In fact, the estimated saving in investment by using high pressure steam in the tube and fin type radiation versus low pressure steam in cast iron radiation is about 20 per cent.

With steam for heating available at 50 lb pressure, it is possible to control the heating system by varying the pressure in accordance with weather conditions. There is situated in the boiler room a pressure recorder control instrument which governs the remote air operated reducing valve. By this means, the operating engineer can vary the heating system pressure at will, using the minimum in mild weather and the maximum pressure during extremely cold weather. Another objective achieved by the scheme is that the type



This installation of three turbo-compressors and carburetors furnishes the gas mixture for heating flaking rolls on the floor above. Also shown is the type of radiation used for heating the plant with high pressure steam



One of the propeller type unit heaters carrying steam at 50 lb (under maximum conditions) which are installed on lower floors

of radiation used is thought to be more adaptable to proper cleaning in dusty locations than more conventional wall type radiation.

A check of the system by the resident engineer indicated that when similar conditions exist this is an economical and satisfactory way of handling industrial heating. In fact, at the new coffee plant General Foods Corp. is erecting at Hoboken, N. J., there is one building five stories high with a ground area of approximately 35,000 sq ft and another building four stories high with a ground area of approximately 10,000 sq ft. both of which have continuous sash on most of the floors. It was decided to carry out the same idea as used at Kankakee for both of these buildings. There are some other structures in connection with the Hoboken project which do not lend themselves to this particular method and are to be heated in accordance with the plan that suits them best.

Process Steam and Piping

A pressure reducing station similar to that for the heating steam, except that its capacity is 750 lb per hr and it reduces the boiler pressure to 5 lb, supplies the hot water heater. A 150/50 lb reducing station of 100 lb per hr capacity furnishes steam to the bag fumigating room, where returned bags are subjected to high temperatures for several hours to sterilize them. This bag room is heated by ceiling mounted radiators of the same type as used throughout the general plant areas, and studies are being made of how best to stack or hang the bags to get proper air circulation throughout them.

There are also pressure reducing stations for supplying the various cookers, steamers and corn oil expellers. For the grits dryer, steam at pressures ranging from 5 to 50 lb, as best suited to operation, is furnished the fin coil through a control valve governed by electric thermostat in the discharge air.

Pipe is of black steel for the steam, return, corn oil, and gas lines, wrought iron with galvanized malleable fittings being used for the cold and hot water piping. Fittings for the 50 lb steam lines are cast iron, steam pattern, 125 lb pressure, and for the 150 lb lines are 250 lb extra heavy cast iron. Lines 2½ in. and larger were welded, and 2 in. and smaller were threaded.

It was specified that all pipe sleeves should be packed with asbestos rope packing to make them dust tight.

P. P. Pratt, western division engineer for General Foods, was in charge of laying out the production equipment, which is of the most modern type and arranged to take advantage of straight gravity flow; drives are by means of totally enclosed motors for minimizing explosion hazards, and the plant is equipped with an extensive dust removal system. E. J. Grimmett was the resident engineer for General Foods during construction, and the entire project was under the executive supervision of R. R. Thomson, General Foods manager of western plants. J. J. Mullen is general manager of the mill.

The mill was designed, built and equipped by The H. K. Ferguson Co. in conjunction with engineers of the General Foods Corp. The heating and piping installation was made by Gallaher & Speck, Inc.

Check Charts for Air Conditioning Calculations

By J. M. Dean*

A GENERAL introduction to this series of charts appeared in the May, 1938, H. P. & A. C. Charts 9 and 10 are reproduced on the next page. Chart 9 is for determining the total pounds of air to be supplied, and Chart 10 is for checking the characteristics of air entering the fan.

These charts were developed in connection with the author's duties as an air conditioning engineer for an electric power company in an endeavor to reduce the work of checking air conditioning jobs to a simple routine procedure.

*Air Conditioning Engineer, Des Moines Electric Light Co. All rights reserved.

Check Charts for Air Conditioning Calculations

[See preceding page]

Chart No. 9

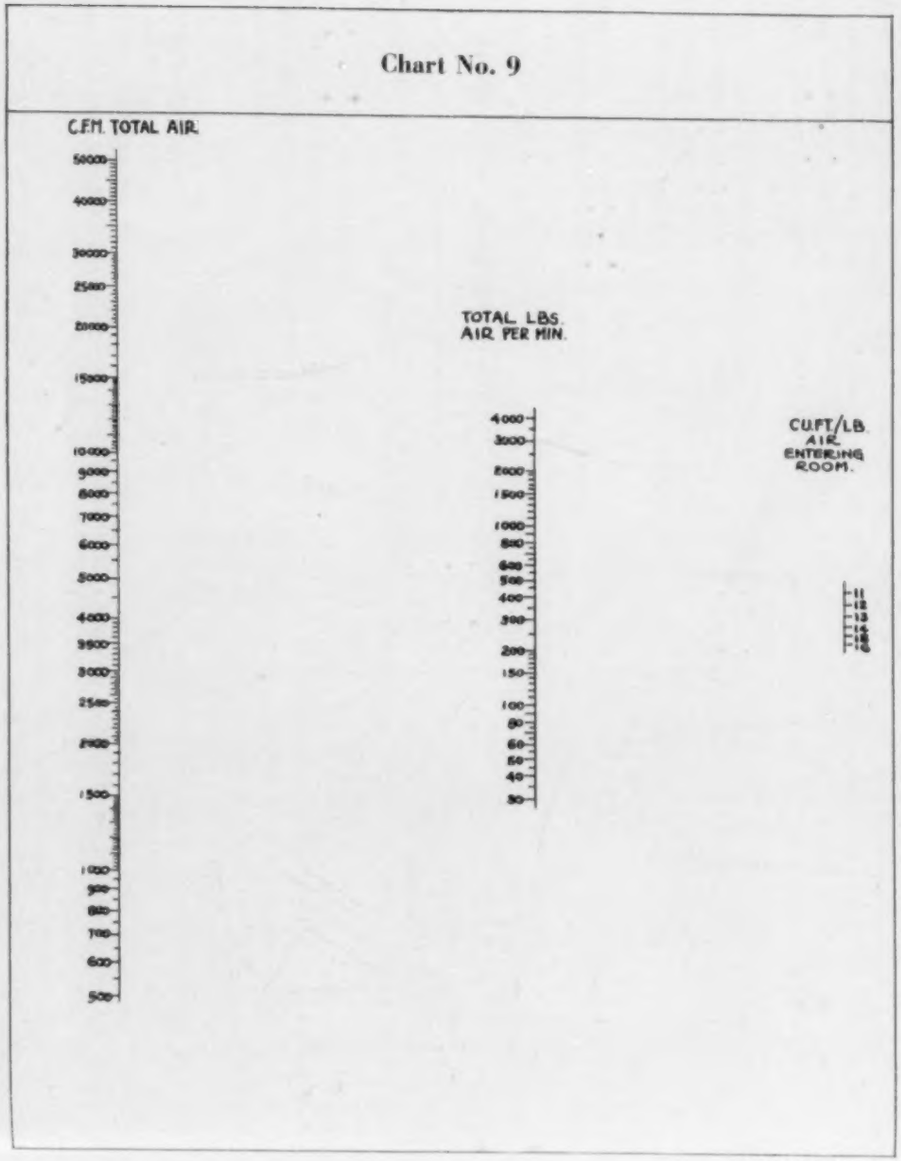
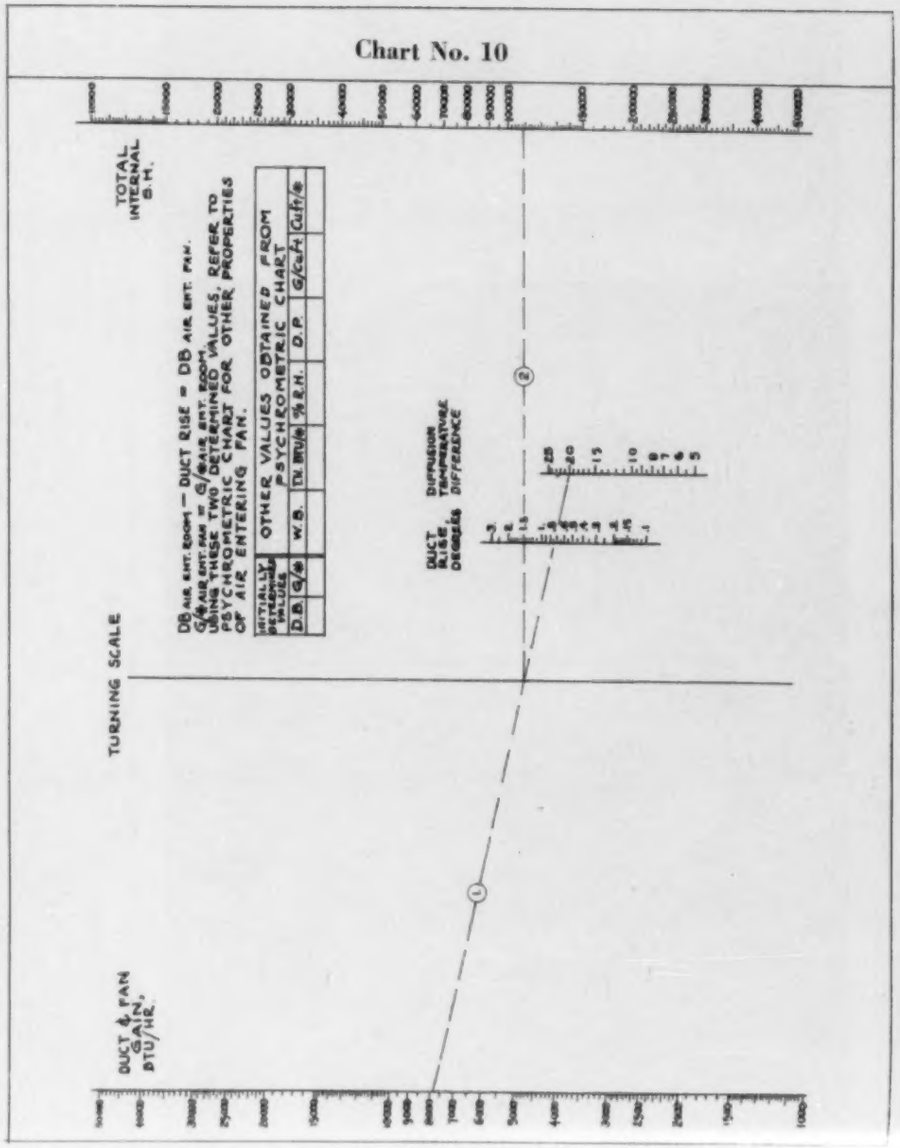


Chart No. 10


TOTAL INTERNAL B.H.
DB AIR ENT. ROOM — DUCT RISE = DB AIR ENT. FAN.
G/# AIR ENT. FAN = G/# AIR ENT. ROOM
USING THESE TWO DETERMINED VALUES, REFER TO PSYCHROMETRIC CHART FOR OTHER PROPERTIES OF AIR ENTERING FAN.
INITIALLY DETERMINED VALUES
D.B.
G/#
OTHER VALUES OBTAINED FROM PSYCHROMETRIC CHART
W.B.
TH. BTU/#
% R.H.
D.P.
G/cu.ft.
Cu.ft/#
DUCT RISE, DEGREES
DIFFUSION TEMPERATURE DIFFERENCE
TURNING SCALE
DUCT & FAN GAIN, BTU/HR.

Atmospheric Pollution in a City and What It Means to the Engineer

By A. D. Singh*, Frank A. Chambers**, and I. A. Deutch***

The method of making an extensive study of atmospheric pollution in a city is described, and the results of the investigation in the various areas as well as in an air conditioned office building are summarized. . . . The authors conclude with a few remarks on the practical significance of the sulphur dioxide content of the atmosphere and the corrosion of metal surfaces

BECAUSE of large quantities of coal containing 0.7 to 5 per cent sulphur consumed in the Chicago territory, the pollution of its atmosphere with the resulting oxides of sulphur has been studied. Zones of high and average pollution with sulphur dioxide have been determined, and an attempt has been made to locate various sources causing this pollution. A discussion of the importance to the engineer of the findings is included.

The midwestern coals are of the bituminous type, containing 35-40 per cent volatile matter, and their sulphur content varies between 0.7 and 5 per cent. On the average 50 to 90 per cent of this sulphur issues from the stacks as oxides of sulphur, depending on conditions of firing. Of the sulphur leaving the chimneys, about 95 per cent is in the form of SO_2 and the remainder is SO_3. Coupling the fact of high sulphur content of these coals with the report that almost 25,000,000 tons are consumed annually in Chicago and vicinity, it becomes of prime importance to determine the concentration of sulphur dioxide in the atmosphere. Consequently, during the first four months of 1937, an extensive survey of SO_2 pollution of the atmosphere was made jointly by the city department of smoke inspection and abatement and the chemical engineering division of the engineering experiment station at the University of Illinois.

Apparatus and Method Used for Survey

The apparatus is shown in Fig. 1. It was mounted in an automobile truck body. The air sample was drawn by tapping a connection into the intake manifold of the automobile engine. It was drawn directly into an alkaline solution of hydrogen peroxide, held in the sintered glass absorption bottle. The sulphur dioxide was readily oxidized by the hydrogen peroxide to sulphuric acid, which was neutralized by the alkali. The supply of air to the peroxide solution was terminated while it was still somewhat on the alkaline side. Excess alkali was back titrated with standard hydrochloric acid. The difference between the standard alkali solution originally taken and that equivalent to back titration with a standardized acid solution represented the quantity of sulphur dioxide present in the air sample drawn through the solution.

In order to study the existing pollution, the city was divided into the following areas: (1) Central business district, generally called the "loop," and vicinity. (2) Residential districts. (3) Manufacturing areas. (4) Railroad stations and locomotive roundhouses. (5) Outlying typical residential districts. (6) Outlying typical manufacturing areas. (7) Stockyards.

1. *The Loop and Vicinity.*—In this section it was found that the concentrations averaged between 0.2 to 0.5 parts per million (ppm). An attempt was made to ascertain how much of the loop pollution came from outside sources. Various winds were traced for several miles, in each case terminating in tests in the loop. On the north and northeast winds, concentrations encountered outside the loop were from traces to 0.1 ppm. Small sources at times tended to raise the concentrations to about 0.2 ppm in certain neighborhoods. In general, these winds brought in less than 25 per cent of the total pollution existing in the loop. The westerly winds were responsible for somewhat higher pollution of the loop. The south and southeast winds were traced from the city limits to the heart of the loop. Tests at the city limits indicated that only a trace to a maximum of 0.05 ppm of SO_2 were coming in from outside sources. As the tests progressed northward, it was found that in the vicinity of 40th Street, concentrations as high as 0.5 ppm could be detected. Traveling north to Roosevelt Rd., these concentrations had undergone considerable dilution so that the average detectable amount did not exceed 0.15 ppm. The tests that followed these in the loop proper indicated concentrations of 0.3 to 0.5 ppm SO_2. On an east wind, there is no polluting factor. Generally then, it may safely be concluded that an appreciable portion of the SO_2 to be found in the loop originates in the same area.

2. *Residential Districts.*—Concerning the residential districts, it was determined that on the North Side the average concentrations varied between 0.1 to 0.15 ppm. On the West Side average concentrations were about the same except in localities where some industrial operations were being carried out. In such cases the concentrations varied between 0.2 and 0.3 ppm. On the South and Southeast, comparatively low concentrations were encountered except between 22nd and 63rd Streets, where concentrations almost comparable to those in the

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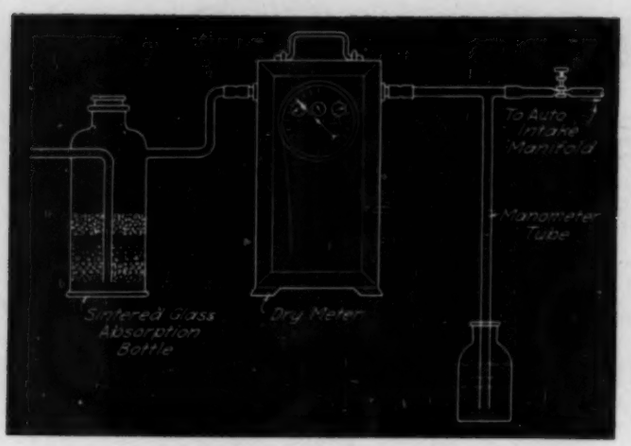


Fig. 1—Apparatus for making tests

loop were found. This was due to population density and the low grade fuels burned.

3. *Manufacturing Areas.*—There are three types of manufacturing districts in the city: (a) Congested districts containing exclusively manufacturing plants. (b) Districts adjoining railroad tracks, canals and other transportation facilities in various parts of the city. (c) Single large plants engaged in various pursuits of industrial activity.

Except on foggy or gusty days the concentrations of sulphur dioxide in the manufacturing areas were lower than those prevailing in the loop. On foggy days, the high relative humidity allows the heavier chimney gases to descend more readily, and on gusty days these gases are blown down repeatedly. Under such weather conditions, on the leeward side of certain plants concentrations as high as 0.8 to 1.0 ppm were detected.

4. *Railroad Stations and Locomotive Roundhouses.*—In a certain railroad station an average concentration of 2.12 ppm was found. In the vicinity of roundhouses, the prevailing concentration was 0.67 ppm and in some cases was as high as 1.6 ppm. Only in one instance the pollution was 0.17 ppm. One noteworthy feature about pollution due to locomotives was that these high concentrations prevailed every day regardless of weather conditions.

5. *Outlying Residential Districts.*—The concentrations encountered in these areas were negligible.

6. *Outlying Manufacturing Areas.*—In these areas considerable variation in pollution was observed. For instance, around steel mills the concentration averaged only 0.05 ppm. On the other hand, one industrial plant gave readings of 1.46 and 1.94 ppm. In Whiting, Ind., an acid concentrating plant emitted a fog containing SO_3 fumes which was very irritating to the nose and throat. Expressed in terms of SO_2, the concentration of trioxide ranged from 0.23 to 0.88 ppm.

7. *Stockyards.*—In the yards proper, from a trace to a maximum of 0.1 ppm of sulphur dioxide was found. Over the territory surrounding the stockyards, the concentrations were 0.2 ppm or more. The sudden disappearance of SO_2 in the stockyards was probably due to its neutralizaion by the basic constituents in the atmosphere created by the decomposition of the organic matter.

Conditioning of Air Containing Sulphur Dioxide

Numerous tests were made in an air conditioned office building on Michigan Avenue close to the Chicago River. Two interesting factors brought forth in these experiments were:

(a) The individual air conditioning units used in most of the offices of this building reduce the concentration of SO_2 in such

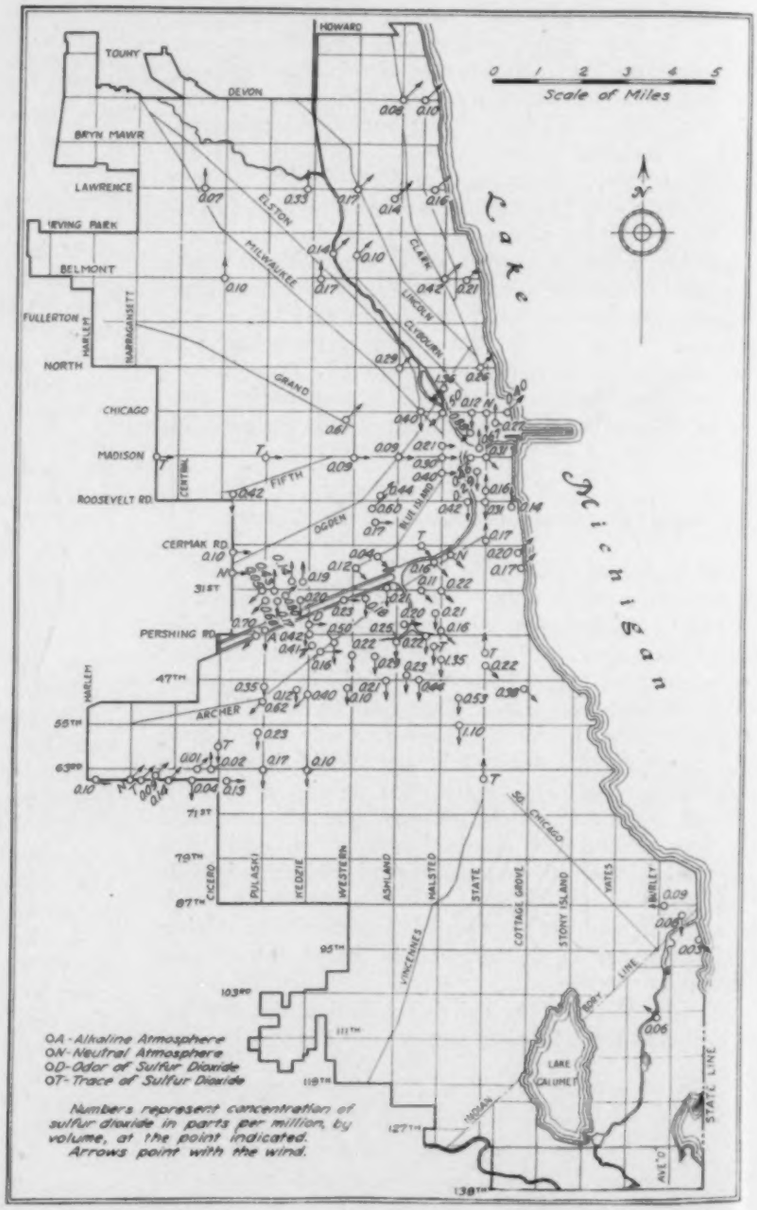


Fig. 2—Distribution of sulphur dioxide in Chicago's atmosphere with various prevailing winds

offices to less than 40 per cent of what it was without the units in operation.

(b) An addition to this building has a central air conditioning unit serving a complete radio broadcasting station including many studios. One of the ducts in this building had to be replaced after six months of operation due to excessive corrosion. However, treatment of the wash water with an alkaline agent properly buffered eliminated the corrosion difficulty.

Effects of Atmospheric Pollution

From the foregoing observations, it is concluded that sulphur dioxide is present in Chicago's atmosphere in measurable quantities. Whether or not the concentrations encountered are definitely injurious to health, vegetation, and property is still very much in the undecided stage. The engineer must, however, keep in view certain points of practical significance. For instance,

(a) Wherever large volumes of air are constantly brought in contact with a moist surface, corrosion difficulties due to SO_2 may be anticipated.

(b) Damage to brickwork and stone construction is not very noticeable. This conclusion applies even to places with the highest concentrations.

(c) In one or two places, samples of marble have been found to be pitted somewhat. Whether or not this pitting is definitely due to sulphur dioxide is not yet established.

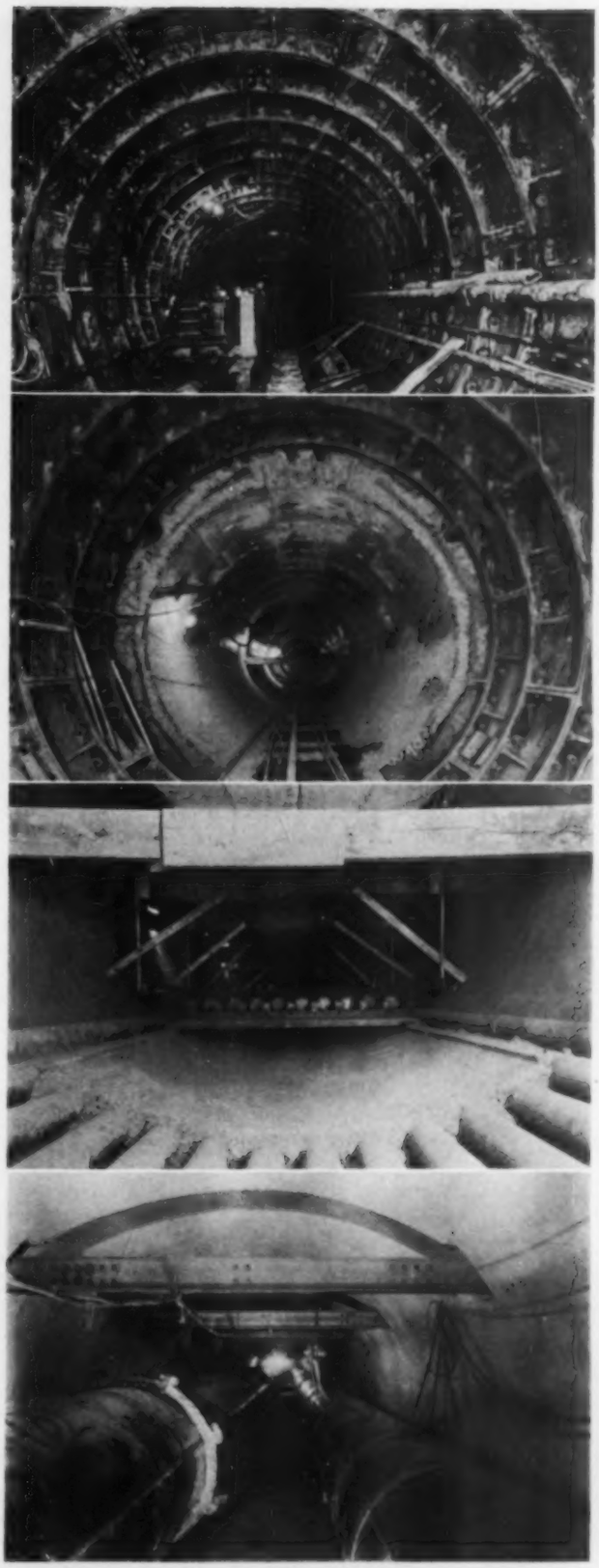
(d) Moist SO_2 will definitely corrode metal surfaces unless they are properly protected with a film of paint or other suitable materials. In regard to the corrosion of metals, one should also consider the part played by carbonic acid and oxygen in the presence of moisture. These constituents are present in the atmosphere in much higher concentrations than is sulphur dioxide. A chemical analysis of rust should reveal the cause of corrosion resulting from various agents in the atmosphere. The city's smoke department is devoting attention to this question at the present time.

Big Tunnel Carries Piping Under Slip

ONE of the most interesting phases of the underground construction work at the Ford Motor Co.'s Rouge plant, Dearborn, Mich., is the service tunnel beneath the Rouge River slip. This 620 ft tunnel, 12 ft in diameter, was installed 70 ft below ground level by the shield method, using cast iron segment liner plates and an inner lining of concrete 18 in. thick. The tunnel carries a 48 in. gas main, 36 in. mill water line, 12 in. city water and 10 in. fire mains, a 16 in. high pressure steam line, 6 in. condensate return line, 10 in. high pressure air line, an 8 in. tar line, and an 8 in. fuel oil line. There are twenty-four 5 in. electrical conduits uniquely embedded in the floor.

The greatest under-surface structure at the Rouge plant, however, is the 2½ mile tunnel, which has been in service for several years, used to bring mill water from the Detroit River. This great underground waterway, with a diameter of 15 ft, carries approximately a billion gallons of water a day, primarily for cooling the 110,000 kw generators in the power house, after which it is used for mill service throughout the plant.

Controlling the intricate system of water services at Rouge are over 1100 valves underground ranging in size from 4 to 54 in., besides the hundreds of valves on many miles of smaller branch lines throughout the buildings.



Top to Bottom—The service tunnel in an early stage of construction. Cast iron sections comprising the tunnel framework were laid in the shaft which had been driven by "sand-hogs," employing the shield method. . . . The cast iron segments were lined with concrete. . . Electrical conduits were embedded in concrete in the floor of the tunnel. . . . The completed tunnel, with larger service lines installed

Air Conditions Discussed at Therapy Congress

THE increasing importance that the air conditioning industry is attaching to cooperation with the medical profession has been mentioned in these pages on a number of occasions. That the "feeling is mutual" and that doctors are more and more appreciating the effects of natural and artificially controlled air conditions in their work is also evident, judging by the annual convention and exhibit of the American Congress of Physical Therapy held in the Palmer House, Chicago, last month. At the previous physical therapy convention there was but one exhibit pertaining to climate and air conditioning; this year there were perhaps half a dozen which were pertinent to the air conditioning engineer's work.

One booth featured detailed plans and a diorama view of a proposed air conditioned "Climatorium" by Dr. I. Goldmerstein of Paris and Prof. K. Stoedieck of Berlin originally designed for the Paris exposition and which is now suggested as a feature for the world's fair to be held in New York next summer. It is a large domed structure intended as an artificial spa for big cities to provide all those benefits of climate throughout the year which now must be sought at expensive resorts. Essentially it comprises a vast beach of warmed sand surrounding a large circular swimming pool; with electrical equipment for the artificial production of sun rays; an atmosphere of pure and healthy air maintained by the air conditioning equipment; and various baths—mud, mineral, air—and mineral waters of all kinds; as well as places for sports and games.

In the adjoining booth, Dr. Madge C. L. McGuinness of New York—who has for some years been concerned with man's air environment—had an exhibit on the effect of climate and health and the advantages of "making climate to order." It is Dr. McGuinness' conviction that every city should have air conditioned facilities where the doctor can specify the weather for his patient just as he can now specify other phases of the necessary treatment.

Frederick Sargent of Phillips Exeter Academy presented an exhibit on weather changes and the onset of colds. By means of charts he showed that the onset of colds in normal preparatory school students is associated with the passage of the so-called "polar front," and that a large number of colds occur with the beginning of this atmospheric change and a small number follow the barometric crest. Dr. George Milles of the University of Illinois college of medicine exhibited data on the relationship of meteorological fluctuations to the clinical occurrence of hemorrhage and perforation in peptic ulcer.

Among the papers presented at the convention sessions was a discussion by Harvard's Dr. Michel Pijoan on the biological effects of therapeutic fever. Research has established optimum conditions for the use of artificial fever therapy, which was introduced to the medical world several years ago as a method of treatment in numerous infections and other conditions. The key to its successful use, according to Dr. Pijoan, is the relative humidity within the cabinet, which should be around 95 per cent. In the treatment, the patient suffers a severe loss of water and numerous chemicals—especially chlorides—due to perspiration. As the heat rises, the patient pants, which causes also a loss of carbon dioxide—which all results in setting up an alkaline condition. If the humidity within the cabinet is kept at a high point (tests were made over a range from 40 to 100 per cent) it is necessary to raise the temperature within the cabinet to but 115 F to bring the patient up to 106 instead of 140 F; this minimizes the chemical loss and avoids panting.

Dr. Charles Sheard, Dr. Marvin M. D. Williams, and Dr. B. T. Horton (of the Mayo Foundation and the Mayo Clinic) presented a paper on Vasomotor Regulation of the Temperature of the Extremities in Health and Diseases, and exhibited charts and diagrams of the studies they have been making on the subject. It will be recalled that a paper on this work by Dr. Sheard and Dr. Williams was presented at the annual meeting of the American Society of Heating and Ventilating Engineers in January, 1937, and was published in the *A.S.H.V.E. Journal Section*, HEATING, PIPING AND AIR CONDITIONING, March, 1937.

Heating and Air Conditioning Indebted to Horticulture

The mechanical arts owe a direct tribute to improvements that were first made in the culture of living things—particularly in horticulture—Lewis Mumford has written in "The Culture of Cities" (Harcourt, Brace & Co.). Early improvements in heating were originally the conception of gardeners, according to Mr. Mumford.

"One of the first persons to suggest steam heating was Sir Hugh Platt, who had the notion of conveying heat from a steam boiler on a kitchen stove through pipes to growing plants in order to keep them at a temperate heat, no matter what the conditions of the weather outside. In 1745 Platt's suggestion was improved by Sir William Cook, who published a diagram for heating all the rooms in the house from the kitchen fire. The full importance of these inventions was grasped for the first time by Joseph Paxton, designer of the Crystal Palace for the industrial exhibition held in London in 1851. In a letter to the *Illustrated London News* (July 5, 1851) he sketched out the design of a Crystal Sanitorium: not merely to give patients the benefit of extra oxygen from the growing plants, but for the sake of sunlight and room wherein to exercise in all weather. To make this possible, he suggested the installation of apparatus to provide filtered and heated air: the first proposal for complete air conditioning—not as a mere remedy against baneful gases, as in mines or in the British House of Parliament, but as a positive aid to health."

Identification of Piping Systems

IT is not as generally known as it should be that there is available a scheme for the identification of piping systems which is approved by the American Standards Association and was originally issued some years ago.[1] Because readers of H.P. & A.C. from time to time inquire for information on this subject, the scheme is being published here with the permission of the A.S.A., although it appeared before (soon after it was first brought out) in our July, 1929, number.

The principal requirements for a standard scheme for the identification of piping systems are distinguishability, flexibility, inclusiveness, simplicity, practicality, and rationality. Its first and principal requirement is that the pipes in each system shall be so colored as to designate exactly which of the five major classes of fluids are being conveyed by the system.

This scheme provides that the color may be applied over the entire length of the piping system or in bands 8 to 10 in. wide near valves and at other important places. The use of bands permits of identification by color, even though all the pipes of a given layout are given uniform treatment for purposes of illumination or decorative appearance.

Where it is desired to identify more exactly the nature of the fluids in a given piping system, this may be accomplished by the addition of one or more stripes of color at the edges of the band, or by stenciled letters, words or figures on the band. For example: in a plant having several safe materials, there would be several pipes having the dominant color or color band of green, and superimposed on these green bands on one piping system might be a stripe of red, on another a stripe of yellow, another a stripe of blue, etc., the color of the secondary stripe having lost its main classification significance under this scheme. If the selected system had included legends instead of stripes, then in the example given above there would be superimposed upon each green band the stenciled name of the material conveyed by that particular pipe, or an identification mark of some kind.

The use of multi-colored schemes in the past has conveyed the general impression that there are numerous colors from which to select and upon which to build a system of identification so wide in scope as to embrace all materials necessary in any plant. In reality, however, colors which show a sufficient difference to be readily recognized under adverse conditions are so few as to make such a scheme difficult to handle. Further, there are comparatively few pigments which may be considered sufficiently permanent to stand up satisfactorily under all conditions of use and exposure. Most of these are high-priced and not in all instances, readily available.

The need is for a clear, concise scheme which will quickly establish in the minds of plant employees the danger or safety of materials carried in a given pipe. Then by the use of secondary color or legend the scheme may be expanded in a given plant or industry to permit the complete identification of the materials in the piping systems.

[1] "Scheme for the Identification of Piping Sysetms," approved by American Standards Association, 29 W. 39th St., New York, N. Y. Price, 50c.

Scheme For the Identification of Piping Systems

Object and Scope

1. The scheme is intended to harmonize all specifications for the identification of materials conveyed in piping systems and is intended to form an acceptable basis for a universal scheme.

2. This scheme has been limited to the identification of piping systems in industrial plants and power plants, not including pipes buried in the ground and electric conduits.

Definitions

3. Piping Systems—For the purpose of this scheme, piping systems shall include in addition to pipes of any kind: fittings, valves, and pipe coverings. Supports, brackets, or other accessories are specifically excluded from applications of this standard. Pipes are defined as conduits for the transport of gases, liquids, semi-liquids, or plastics, but not solids carried in air or gas.

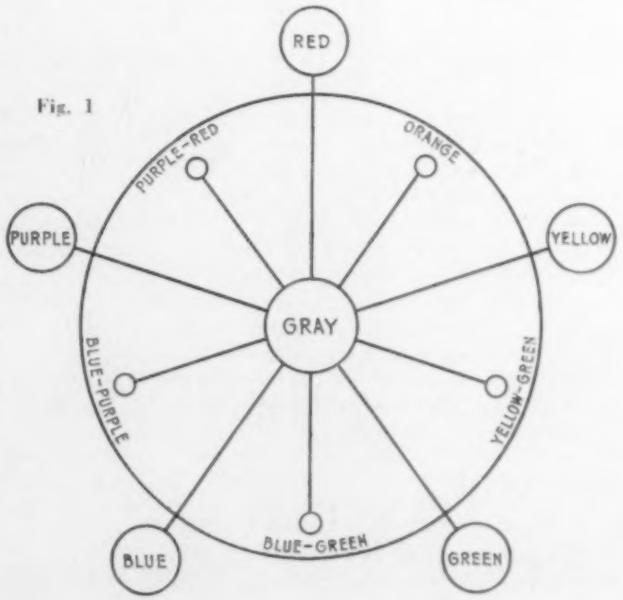
4. Fire Protection, Materials, and Equipment—This classification includes sprinkler systems and other fire fighting equipment.

5. Dangerous Materials—These materials are those which are in themselves hazardous to life or property by virtue of being easily ignited or productive of poisonous gases or are in themselves poisonous. They include materials that are known ordinarily as fire producers and explosives.

6. Safe Materials—These products are those involving no hazard in their handling and no extraordinarily high value. A workman in approaching a piping system to make repairs will accordingly run no undue risks in breaking into a pipe bearing a safe material, even though that material had not been emptied by previous arrangement.

7. Protective Materials—Under this class fall materials which are piped through plants for the express purposes of being available to prevent or minimize the hazard of the dangerous materials above mentioned. Thus, a plant may have certain special gases which are antidotes to poison fumes, piped through its shops for the express purpose of release in case of danger.

Fig. 1



8. Extra Valuable Materials—These might be classified with the group of the safe materials above mentioned, but where these products have a very high value, it is preferable to give them a separate major classification.

Method of Identification

9. At conspicuous places throughout a piping system color bands shall be painted on the pipes to designate to which one of the five main classes its contents belongs. If desired the entire length of the piping system may be painted the main classification color. (See paragraph 13.)

10. Further, the actual contents of a piping system may be indicated by, preferably, a stenciled legend giving the name of the contents in full or abbreviated form. These legends shall be placed on the colored bands. The identification scheme may be extended by the use of colored stripes placed at the edges of the colored bands.

11. The bands, legends, and stripes shall be placed at intervals throughout the piping system, preferably adjacent to valves and fittings to ensure ready recognition during operation, repairs and at times of emergency.

Main Classification by Color

12. This part of the scheme is intended to identify the main classes into which the materials in the piping systems belong. It establishes a basic fundamental principle applicable to all schemes for identifying piping systems and is intended to designate the pipe contents as belonging to a specific class of materials that are safe, dangerous or otherwise, depending on local conditions. The inclusion of all existing systems at present in use is made readily possible with minimum change by the adoption of the following color classification.

13. All piping systems are classified according to the character of material carried. Especially in an emergency the quick recognition of the contents of a piping system is of paramount importance. For this purpose, each piping system is classified, by the nature of its contents, into the following classifications:

Class	Color
F—Fire protection equipment	Red
D—Dangerous materials.	Yellow (or orange)
S—Safe materials	Green (or the achromatic colors, white, black, gray, or aluminum)
P—Protective materials.	Bright blue
and, when required	
V—Extra valuable materials	Deep purple

14. The above colors have been chosen to identify the main classification because they are readily distinguishable one from another. Reference to Appendix B will show in addition the relation of the colors chosen to each other and to the color circle on which the color arrangement has been based.

15. Red has been assigned to fire protection equipment because of long established custom. Yellow and orange have been assigned to dangerous materials because of all the saturated chromatic colors these have the highest coefficient of reflection under white light and can therefore be more readily recognized under the poorest conditions of illumination. Further, yellow is the traditional color of the quarantine flag and has been adopted as the caution signal for railroads and road traffic. The assignment of the other three colors follows in natural order.

Detailed Instructions for Working Out a Standard Identification Scheme[2]

16. List all materials carried in the pipes of the system or systems.

17. Assign each of these materials to one of the five main classes: Safe products (S), Dangerous materials (D), Protective materials (P), Extra valuable materials (V), and Fire protection equipment (F).

18. Group the materials assigned to each class for the purpose of facilitating the selection of sub-class markings.

19. Choose between the alternative methods of (a) color bands or (b) complete color painting.

20. Assign a legend or color stripe to each material listed under each main classification.

21. Refer to Appendix C and select the five colored paints which will identify the main classes and the paints to be used for the stripes if the use of stripes has been decided upon in place of legends.

22. No type of paint or other suitably colored coating is excluded. The selection of coatings comes under the jurisdiction of the supervising engineer or person in authority and may be made to meet the diverse requirements of each case. All that the scheme requires is that the dominant hue of the band shall fall unmistakably within one or the other of the five spectral regions.

23. Colors which have a total reflective value too low to permit ready differentiation at minimum illumination are excluded. Median gray, which from the physiological or visual standpoint is situated about halfway between white and black, reflects approximately 25 per cent of the incident standard white light and is near the permissible limit of reflective value for poor illumination. The permissible limit of reflective value is provisionally established at 19 per cent.

24. An ordinary hand flash lamp when held by an operator at the reading distance may be used as the arbitrary measure of minimum illumination.

[2]The Sectional Committee on the Identification of Piping Systems has created a permanent organization to enable it to assist those firms and individuals who desire to consult it preparatory to the installation of this standard scheme.

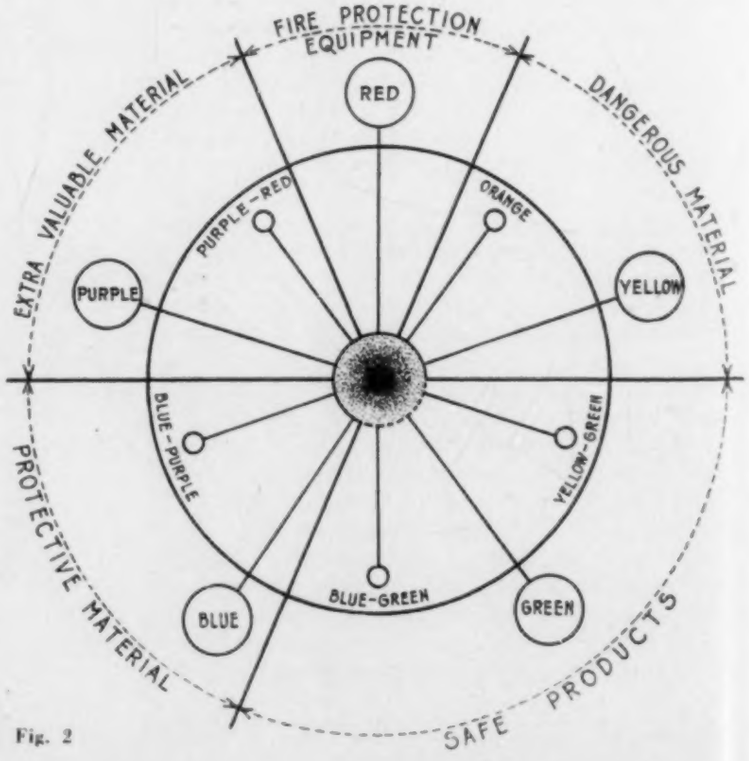


Fig. 2

Classification of Materials Carried in Pipes

Material Piped	Physical State Gas, Liquid, or Semi-Solid	Temperature of Material in Deg Fahr (Max.)	Pressure in Lb per Sq In. (Min.)	or Vacuum in Inches of Mercury or Water	Classification
Acetic acid	Liquid	Normal	30 lb	28 in. Hg	D
Alcohol	Liquid	Cool			D
Amyl acetate	Liquid	Cool			D
Alum solution	Liquid				S
Acetylene gas	Gas	0 to 200	½ in. to 250 lb		D
Benzol	Liquid	Cold	Up to 80 lb		D
Brine	Liquid	Cold	About 60 lb		S
Butyl alcohol	Liquid	Cool			D
Bleach liquor	Solution				D
Bisulphite liquors	Liquid				D
Blau gas	Gas	00 to 200	½ in. to 250 lb		D
Compressed air	Gas	80	300 (in special cases 3000 lb)		S
CO_2	Gas and liquid	— 30 to — 100	0 to 250 lb per sq in. ga		D
Caustic soda solution	Solution				D
Chlorine	Gas, liquids, and solutions				D
Carbon bisulphide	Gas and liquid				D
Chloroform	Gas and liquid				D
Carbon tetrachloride	Liquid				S
Coal gas	Gas	0 to 200	½ in. to 250 lb		D
Dyes	Solution	Usually cool			D
Flue gas or waste heat	Gas	0 to 200	½ in. to 250 lb		D
Fuamite	Liquid	Cold	Up to 100 lb		F
Glycerine	Liquid	Hot and cold	Up to 60 lb		S
HCl	Liquid	Up to 80	2 to 10 lb		D
Hydrogen	Gas				D
H_2S Gas	Gas	0 to 150	½ in. to 100 lb		D
Intermediates	Solution	Usually cool			D
Lactic acid	Liquid	Normal	60 lb	20 to 22 in.	V
Mixed acid	Liquid	Normal	30 to 80 lb		D
Miscellaneous solvent	Liquid	Cool			D
Mercury	Gas and liquid		29 in. to 100 lb		D
Muriatic acid	Liquid				D
Natural gas	Gas	0 to 200	½ in. to 250 lb		D
Nitric acid	Liquid	Normal	10 lb		D
Nitro bodies	Liquid	Cool	Up to 40 lb		D
NH_4NO_3 (molten)	Liquid		None		D
NH_3	Liquid and gas	— 30 to 100	0 to 250 lb per sq in. ga		D
N_2O and N_2O_3 vapor	Vapor	0 to 150	½ in. to 100 lb		D
Oils (Petroleum)	Liquid	Below 580	Below 1200 lb		D
Paper sizing solution	Liquid				S
Pinch gas	Gas	0 to 200	½ in. to 250 lb		D
Producer gas	Gas	0 to 200	½ in. to 250 lb		D
Pyrox compounds	Liquid	Up to 80			D
Paint	Liquid	Cold	30 lb		D
H_2SO_4	Liquid	Up to 180	30 to 80 lb		D
Soda ash solution	Liquid	Cold	60 lb		S
Sugar juices and syrups	Liquid	195 to 200	0 to 50 lb		S
Steam	Vapor	Below 212	Below atmos.		S
Steam	Vapor	212 to 800	Above atmos.		D
SO_2	Gas and liquid	— 30 to 100	0 to 250 per sq in. ga		D
Sulphur chloride	Gas and liquid				D
Turpentine	Liquid	Cold	Up to 80 lb		D
Toluene	Liquid	Cool	Up to 40 lb		D
Tar	Liquid				D
Varnish		Cold and hot	30 lb		D
Vapors of valuable materials not otherwise specified	Vapor	0 to 180	½ in. to 100 lb		D
Water	Liquid	Cold	Any pressure		S
		Hot	Any pressure		D
Water gas	Gas	0 to 200	½ in. to 250 lb		D

Key to Classification: F—Fire Protection—Red. D—Dangerous—Yellow. S—Safe—Green. P—Protective—Blue. V—Extra Valuable—Purple.

Appendix A—Classification of Materials Carried in Pipes

For the purpose of securing data on which to base its report, the sectional committee through its sub-committee No. 3 on classification canvassed the various industries. This canvass was carried through by means of a questionnaire which was sent to a large number of companies typical of the industries handling large quantities of varied kinds of materials which are conveyed in pipes.

In the table which forms part of Appendix A are listed all the materials suggested in the replies to the questionnaire. In addition certain determinants of important physical characteristics are given together with the classification into the five main groups. (S) Safe Products, (D) Dangerous Materials, (P) Protective Materials, (V) Extra Valuable Materials, (F) Fire Protection Equipment.

The committee found that frequently certain materials which were at one time dangerous could fall in one or the other of the classifications. After discussing the matter with representatives of the industries the committee has come to the conclusion that "once a dangerous material aways a dangerous material" should be the rule to be followed, and in the particular industry in which the dangerous material becomes the antidote or safe material, it should still be marked as dangerous. The few men who are in that industry should be directed by markings other than color to use it when necessary for their protection. In this way the hazard of having dangerous materials used on the wrong occasions for safe materials will be eliminated. An example of this is the use of ammonia, usually a dangerous material, as a safety material when attacking phosgene fumes. The contrary case, however, of a normally safe material becoming dangerous under certain conditions, should be met by having this material considered as dangerous only in those locations where it is dangerous, being classified by this committee as a safe material. An example is water over quicklime. In the case of drinking or service water systems with special outlets for fire protection, the committee recommends that the fire outlets only be designated as fire protection, the remainder of the line to be classified as safe.

Appendix B—Identification by Colors

For quite a long time pipe lines in various industrial plants have been painted in different colors for the purpose of ready identification. Of the various color schemes heretofore devised none has been based upon any definite fundamental principle, either scientific, psychologic, esthetic, or traditional. Because the selection of colors has been arbitrary and influenced by local conditions not common to plants in general, and because a comprehensive extension of such schemes soon involves a mass of detail which automatically renders them difficult if not impracticable, the inadvisability of recommending any of them for adoption as a universal scheme is fairly obvious.

This condition was fully recognized by the sub-committee on plan and scope. A part of its report is therefore quoted below:

It is obvious that to attempt to outline a code of identification, would result in a system so comprehensive that even should the supply of colors and identification symbols hold out, adoption of it would automatically be rendered impossible in those industries which do not have a major group of colors allocated to its products. It is found, however, upon investigation that any materials transported in pipes in a plant fall in one of the following classifications:

(a) Safe Products. This represents a majority of the products that are handled through a plant. These products may be defined as having no hazard in their handling and no extraordinarily high value, so that a workman in approaching a piping system to make repairs will run no undue hazard in breaking into a pipe bearing a safe material, even though that material had not been emptied by previous arrangement.

(b) Dangerous Materials. These materials are those which in themselves are hazardous to life or property by virtue of being easily flammable or productive of poisonous gases or are in themselves poisonous. They include of course materials that are known ordinarily as fire producers and explosives.

(c) Protective Materials. Under this class are materials

which are piped through plants for the express purpose of being available to prevent or minimize the hazard of the dangerous materials above mentioned. Thus, a plant may have certain special gases which are antidotes to poison fumes, which gases are piped for the express purpose of affording protection in emergencies.

(d) Extra Valuable Materials. These might be classified as a group of the safe materials above mentioned, but inasmuch as cases came to your committee's attention where those products would have a very high value, it appeared preferable to give them a separate major classification.

(e) Fire Protection Equipment. This might properly be called a division of the Protective Materials just mentioned above, though the hazard of fire and the use of automatic sprinkler systems and other fire fighting equipment having become so universal, it would appear better to make it a special major classification.

These five classifications or subdivisions thereof, if necessary, in the opinion of the committee should each be given a major color, and the various subdivisions that a plant may need can be obtained by the use of numbers, names, or the like, painted in white or black upon a background of the color selected.

The immediate problem of the subcommittee, therefore, was to select a set of colors for these five major classes that would tend to relate, in some suggestive manner, one class with another.

In seeking some common starting point, the solar spectrum was decided upon as embodying a basic principle of color arrangement that is more or less universally understood. When the colors which compose the spectrum are grouped in their natural order in the form of a circle, an arrangement is established similar to that shown in Fig. 1. This color circle may be divided into any number of given parts that may be required, but for purposes of this code, five divisions or sectors, each covering a certain spectral region, have proved quite sufficient. Each of these sectors represents one of five main classes and is capable of subdivision, where necessary, into a larger number of hues having approximately similar dominant wave lengths, but distinguishable one from another very readily under favorable illumination (see Fig. 2). The logical arrangement of the colors shown in this circle enables complementary or opposite colors to be employed as designating a classification of a material directly opposite to one of the other materials appearing in one of the other main classes. For example, orange and blue are complementary or opposite colors and, in consequence, blue is selected as designating the protective materials that would be utilized to minimize hazard where dangerous materials, designated by the complementary orange color, were present. In the same manner, we find green indicative of safe products, as opposite to the complementary red which has always been associated with fire and has, in consequence, been employed to designate fire protection equipment.

With regard to durability there seemed at first to be a general idea that the committee should select five or six indestructible pigments and have paints made from these only. A little reflection soon shows a position of this sort to be untenable, first because the resistance to attack is limited by the vehicle as well as by the pigment, second there are few substances either in the inorganic colored pigments or the organic vehicles that are equally immune to all the various forms of attack, and third the commercial availability of the materials from which practical and economical paints can be made is by no means a negligible factor.

The committee therefore has concentrated its efforts on the presentation of a practical system having a sound claim to preference over any other heretofore suggested, and has avoided confusing the main point at issue with paint production problems which may be easy or difficult of solution according to the differing conditions in different plants. On the other hand the committee has a detailed practical knowledge of what it is possible to produce in paints and has included in the system nothing beyond these possibilities. If a pipe contains extra hazardous material the only fundamental requirement is that it should be yellow; the shade of yellow, the kind of pigment, and type of paint or other coating, are matters of detail coming under the province of the engineer in charge. All the system requires is that the yellow shall be of such a kind that it will not be mistaken for the green, blue, purple, or red of the other main classes.

The system as suggested does not lose sight of an important fact in economical paint production, a fact which has heretofore taken a paramount position in the painting of iron and steel where the protection against corrosion has been of equal or greater importance than the color of the coating. Black, for instance, has been widely used, not only because it may be made at a low price, but also because it is practically the only paint that will withstand elevated temperatures without discoloration, has a relatively longer life under direct exposure to sunlight, and in the form of asphaltic varnishes is proof against all acids, alkalis, and sulphide sulphur compounds even in relatively strong concentrations. It is the ideal paint for pressure steam pipes, but we have taken the position that in most modern plants such pipes are insulated by various kinds of covering mostly white or nearly so, and in general not in need of any kind of paint. Such pipes may still be painted black as protection against corrosion, and the covering marked for identification by the colored bands, legends, and stripes.

The so-called metallic browns and other types of red oxide of iron have also been very largely used in low priced anticorrosive paints. While the reflective value of the higher qualities of Venetian red is 19 per cent, or better, the darker shades run somewhat lower, and for this reason it is believed the dark browns may be excluded from the system with less sacrifice than attends any attempt to retain them.

Gold-bronze powders are of no special value and may be omitted without economic loss, but if included for any special reason would fall in the yellow or dangerous sector. Aluminum paints also are very useful in many cases and have been assigned to be used with the whites and grays to the identification of Safe Products (S).

Appendix C—Description of Colors for Special Application

To assist in the application of the standard scheme for the identification of piping systems the following description list of fourteen colored pigments is given. This list does not cover the entire range of available colors but merely serves to illustrate the means at our disposal for meeting the more usual types of abnormal exposure conditions.

A plant atmosphere that will ordinarily support human life without material danger to general health, may at times be charged with gases or vapors that will react chemically on many paint pigments with a resultant change in color.

By far the most common of these abnormal forms of attack come from vapors that are acidic, alkaline, or sulphureted. The dry gases are far less active but when accompanied by aqueous vapor they penetrate the paint film, and, if reactive with the pigment, cause rupture of the film and change of color by the formation of new compounds having a different volume and color from the original pigment.

If the vehicle is sufficiently resistant it may delay such action for a longer or shorter time according to circumstances, but no paint vehicle now known is wholly impervious to osmotic penetration of this kind. In any plant, therefore, where such abnormal conditions persist, the first step is to use a vehicle having the highest practical water resistance and then select colors in such a way that the pigment will have the least reactivity with the prevailing gas. The colors here described are some of the possibilities in this direction.

Toluidin Red—Fairly fast to aqueous acids, alkalis, and sulphur. An organic pigment that will stand up as well as the vehicle. Adds materially to the life of varnish vehicles and works well with all of them.

Paranitranilin Red—Similar to the toluidin and considerably cheaper, but owing to a slight solubility in the vehicle it has a tendency to stain up through light shades or whites applied over it, e.g., white letters or aluminum stripes.

Red-Orange—A mixed pigment consisting of an organic red

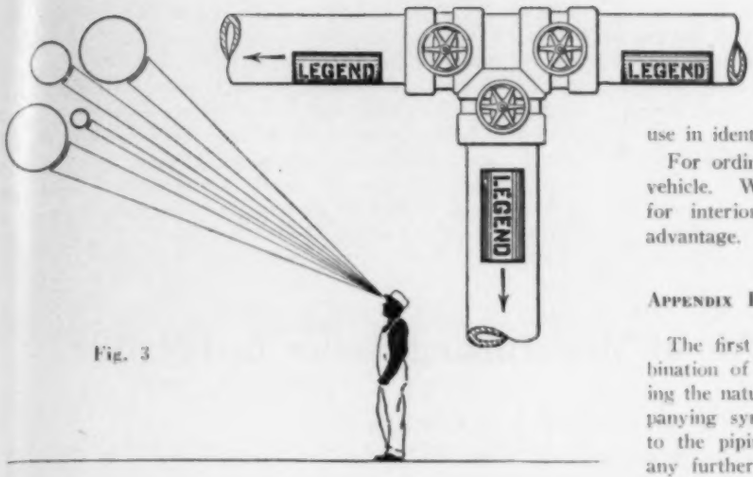


Fig. 3

similar to toluidin mixed with zinc yellow. Fast to alkalis and sulphur but not to acids. Shade of orange may be adjusted to suit. Mixed with cadmium yellow in place of the zinc yellow it becomes fast to acids as well.

Orange-Chrome Yellow—The regular color for ordinary use. Fast to alkalis but not to acids or sulphur.

Yellow—Yellow ochre toned up with bright yellow to a type commonly known as golden ochre. Ochre itself as compared with the more highly saturated hues is little more than a yellow-brown, but in the proposed system where browns as distinctive colors are eliminated the ochre falls unmistakably in the yellow sector. Owing to its general stability and low price it is the main reliance for fast yellow in situations where chrome yellows cannot be used. Like other iron compounds that contain hydroxyl groups or "chemically combined water," it is not wholly fast to sulphur. Strong ammonium sulphide darkens it somewhat but except in extreme cases it is fast to everything. When toned up to a brighter shade, the resistance depends on the kind and quantity of toner.

Cadmium Lithopone—A combination of cadmium sulphide and blanc fixe. Fast to everything, but applicable to extreme cases only because the available supply is limited and the cost relatively high.

Chrome-Yellow-Lemon—The ordinary pigment for general use. It is fast to acids but not to caustic alkali or sulphur.

Green—Chrome oxide toned up with a little yellow to raise its total reflection value. This oxide is a moderately low priced oxide not to be confused with Vert de Guignet, a much more brilliant and expensive quality. It is fast to everything and useful in extreme cases.

Ordinary Chrome-Green Light—Fast to weak acids but not to alkali or sulphur.

Prussian Blue—A tint with a white base of Titanox. It is fast to acid, fairly resistant to sulphur and easily affected by alkali.

Ultramarine Blue—A tint with a white base of zinc oxide. It is fast to alkalis and sulphur but not to acids.

Purple—Base white of Titanox tinted with alizarine purple. Fast to everything. Shade can be adjusted by adding alizarine red in varying proportions.

Gray and Black—The achromatic colors white, black and gray and aluminum are permitted for use in identifying safe products.

For ordinary outside exposure linseed oil is the most durable vehicle. Where higher resistance to water is required and, for interior use, China-wood-oil varnishes may be used to advantage.

Appendix D—Identification of Piping Systems by Legends

The first efforts of the sub-committee centered upon a combination of symbols or hieroglyphics along with a legend naming the nature of the containing fluid, the legend with its accompanying symbol appearing prominently at the points of supply to the piping system and at points of distribution therefrom; any further marking at intermediate points along the pipe line to consist of the symbol only.

The symbol idea was later entirely discarded since it was considered impracticable because of the great mass of detail involved, the difficulty an operating force would encounter in recalling to mind the different shapes and forms and the probable necessity for the publishing of a code book upon the part of each operating company.

The committee finally arrived at the conclusion that this form of identification marking should consist principally of a lettered legend, abbreviated or otherwise, naming the material carried in the pipe line. Where a knowledge of the direction of fluid flow may be of service it is suggested that arrows be painted on the pipe.

Attention has been given to the question of visibility with reference to pipe markings and this committee recommends that where pipe lines are located some distance above the normal line of the operator's vision that the lettering be placed below the horizontal center line of the pipe as shown in the accompanying sketch, Fig. 3. This sketch shows a group of pipes located near a wall. It is desirable, where the view is unobstructed, that the lettering employed be stenciled on the two lower quarters of the pipe covering rather than one side only. The lettering in the position shown will be less liable to be obscured by dust collection or mechanical damage.

In certain types of plants it may be desirable to label the pipes at junction points or points of distribution only, as suggested in Fig. 3. In certain existing power plants the mark-

Outside Diam. of Pipe or Covering, In.	Size of Stencil Letter, In.	Outside Diam. of Pipe or Covering, In.	Size of Stencil Letter, In.
¾	½	5	1½
1	½	6	1¾
1¼	½	7	2
1½	¾	8	2½
2	¾	9	2½
2½	⅞	10	3
3	⅞	11	3
3½	1¼	12	3½
4	1¼	13 and over	3½
4½	1½		

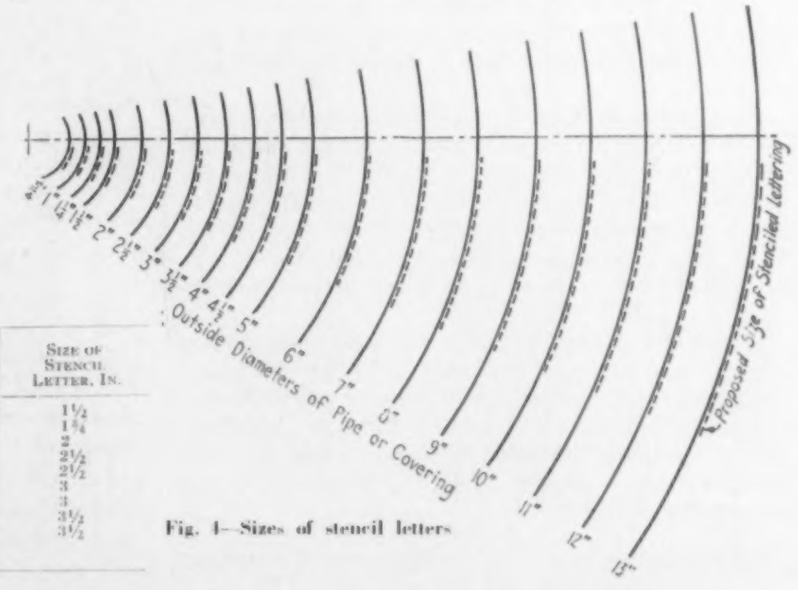


Fig. 4—Sizes of stencil letters

ings are repeated at intervals all along the pipe line. In any case, the operating man should decide the location and number of signs required in each particular system of piping.

Regarding the type and size of letters and numbers, we recommend the use of stencils of standard sizes easily obtained from any stencil manufacturer, ranging in height from ½ in. to 3½ in. For pipes smaller than ¾ in. in diameter requiring identifying marks we suggest the use of a metal tag with the lettering etched and filled in with enamel.

Fig. 4 gives a tabulation and a diagram of suggested heights of letters for different oustide diameters of pipes or pipe coverings.

With regard to visibility, the eye as a matter of habit, is most accustomed to the reading of black letters on lighter backgrounds. We therefore suggest that the black letters be either stenciled or painted on the band of main classification color when light and on a rectangular background when dark. The color of the background and the nature of the material used may be that deemed suitable for the purpose. Aluminum paint serves very well for this purpose.

Industrial Plant Asks About Modernizing Boiler Installation

Samuel R. Lewis* answers H. P. & A. C. reader's question about replacing forty year old plant

A SUBSCRIBER who is president of an eastern manufacturing plant asks where he can find authoritative information as to whether it will pay to change his 200 hp boiler plant from coal burning to oil burning. He states that he burns $7000 worth of bituminous coal costing $7.25 per ton, and that bunker C oil costs $0.025 per gal. The impression he gives is that the boiler is about 40 years old and that the comparison should be made between: (1) Present manually fired old boiler. (2) New boiler with oil burner. (3) New boiler with stoker.

It is of course not practicable without an inspection and possibly a test, and without consideration of other factors such as percentage of rating ordinarily in effect, space in boiler room, and the like, to make a conclusive recommendation. However the following percentages of efficiency are approximately correct:

Over-all efficiency of present plant.....55 per cent
Over-all efficiency with new oil burning boiler. No coal passing or ash removal needed .....75 per cent
Over-all efficiency with new stoker fired boiler. The same labor as at present is likely to be required with coal.....70 per cent

7000 ÷ 7.25 = 966 tons of coal. Probable heat content 12,000 Btu per lb or 24,000,000 per ton, at 55 per cent = 13,200,000 Btu per ton and 12,751,200,000 Btu per year required. This costs at present $0.000549 per 1000 Btu.

Using a new oil burning boiler with bunker C oil having 150,000 Btu per gal and costing 2½c per gal, at 75 per cent efficiency the cost per 1000 Btu would be (0.025 × 1000) ÷ (150,000 × 0.75) = $0.000222. The fuel cost then is (0.000222 ÷ 0.000549) × 100 = 40 per cent of the present cost of fuel. Burning oil it is probable that the cost of at least one fireman will be saved.

With a new stoker fired boiler burning a much poorer grade of coal having a Btu value of 10,500 per lb and costing say $4.50 per ton, at 70 per cent efficiency, the cost per 1000 Btu would be (4.50 × 1000) ÷ (10,500 × 2000 × 0.70) = $0.000306. The fuel cost on this basis is (0.000306 ÷ 0.000549) × 100 = 56 per cent of the present cost of fuel.

The analysis indicates that whether a new oil burning boiler or a new stoker fired boiler is installed, there would be very substantial savings over the present cost. The type of new installation will depend upon the relative cost of oil and stoker coal as well as on the initial investment for the new plant.

It appears that the local price conditions are very much in favor of preheated heavy oil, and that the owner would be wise to employ a consulting engineer.

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

Visiting Firemen Hear Talk on Air Conditioning

Speaking before the International Association of Fire Chiefs in convention at New Orleans last month, L. Logan Lewis, vice-president and chief engineer of Carrier Corp., said that in installing an air conditioning system "there is ample opportunity to choose safe equipment."

Among the fire control measures suggested by Mr. Lewis were elimination of inflammable materials; use of non-combustible filters; periodic cleaning of duct systems; and automatic operation of sprinkler systems to shut down fans after fire breaks out.

"The greatest opportunity for improvement seems to lie in the direction of uniformity of safety requirements throughout the country," he emphasized. "With uniformity of practice, safety devices may be standardized and perfected to a far greater degree and the lower costs which always follow standardization will lead to still better practice."

Return of the Chimney Sweep

He predicted return of the "chimney sweep" in the modern role of a service man for air conditioning. Discussing necessity for clean duct work as a means for fire control, he said air conditioning people are regularly receiving an increased number of requests from owners of systems for names of people who have developed the art of cleaning duct systems. Such people are gradually springing up and developing skill and technique and it is not unexpected that the chimney sweep will soon come back into his own.

Figuring Solar Heat Gains of Buildings

By **William Goodman***

THE first five appendices were published in the September issue. Publication of Appendices 6-9 this month concludes this series of articles on figuring solar heat gains, the first installment of which appeared in the May, 1938, H. P. & A. C.

Appendix 6—Derivation of Equation 10

$$P_{A2} = (1 + P_{T1})\ P_{A1}$$

Referring to Fig. 8, $(I''_1 - I'_2)$ is the radiant energy absorbed by the second plate of glass.†

Equations analogous to [b] and [g] of Appendix 3 can be written for the second thickness of glass alone as follows:

$$I''_1 = I_1(1 - P_F) \quad \ldots\ldots [a]$$

$$I'_2 = \frac{I_2}{1 - P_F} \quad \ldots\ldots [b]$$

Subtract [b] from [a],

$$I''_1 - I'_2 = I_1 (1 - P_F) - \frac{I_2}{1 - P_F}$$

$$\frac{I''_1 - I'_2}{I_1} = (1 - P_F) - \frac{I_2}{I_1}\frac{1}{1 - P_F} \quad \ldots\ldots [c]$$

Eliminate (I_2/I_1) from [c] by means of [a] of Appendix 5, resulting in

$$\frac{I''_1 - I'_2}{I_1} = (1 - P_F) - (1 - P_F)e^{-kL}$$

$$= (1 - P_F)\ (1 - e^{-kL}) \quad \ldots\ldots [d]$$

Now eliminate I_1 from [d] by means of [k] of Appendix 3.

$$\frac{I''_1 - I'_2}{I_u} = (1 - P_F)^2(1 - e^{-kL})\ e^{-kL} \quad \ldots\ldots [e]$$

Now $\dfrac{I''_1 - I'_2}{I_u}$ is the fraction of the incident energy that is absorbed by the second thickness of glass alone. Let

$$P''_{A2} = \frac{I''_1 - I'_2}{I_u} \quad \ldots\ldots [f]$$

Therefore,

$$P''_{A2} = (1 - P_F)^2(1 - e^{-kL})\ e^{-kL} \quad \ldots\ldots [g]$$

The total fraction of the incident energy that is absorbed by the two thicknesses of glass is

$$P_{A1} + P''_{A2} = P_{A2}$$

Now Equation 8 of the text is,

$$P_{A1} = (1 - P_F)\ (1 - e^{-kL}) \quad \ldots\ldots [h]$$

Adding [g] and [h],

$$P_{A2} = (1 - P_F)^2(1 - e^{-kL})\ e^{-kL} + (1 - P_F)\ (1 - e^{-kL}) \quad \ldots [j]$$

$$= [(1 - P_F)^2\ e^{-kL}]\ (1 - P_F)\ (1 - e^{-kL})$$
$$+ (1 - P_F)\ (1 - e^{-kL}) \quad \ldots\ldots [k]$$

But Equation 7 of the text is,

$$P_{T1} = (1 - P_F)^2 e^{-kL}$$

Substituting this equation into [k],

$$P_{A2} = P_{T1}\ (1 - P_F)\ (1 - e^{-kL}) + (1 - P_F)\ (1 - e^{-kL})$$
$$= (1 + P_{T1})\ (1 - P_F)\ (1 - e^{-kL}) \quad \ldots\ldots [l]$$

Now Equation 8 of the text is,

$$P_{A1} = (1 - P_F)\ (1 - e^{-kL}) \quad \ldots\ldots [m]$$

Substituting [m] into [l]

$$P_{A2} = (1 + P_{T1})\ P_{A1}$$

Appendix 7—Computation of P_{B1} and P_{B2} in Table 11

In order to use Equations 7 to 10 inclusive, the value of P_F, the percentage of light reflected from a single boundary surface, must first be computed by means of Fresnel's formula[5] which is

$$P_F = \frac{1}{2}\left[\frac{\sin^2(i - r)}{\sin^2(i + r)} + \frac{\tan^2(i - r)}{\tan^2(i + r)}\right] \quad \ldots\ldots [a]$$

i = angle of incidence.
r = angle of refraction.

The angle of incidence is related to the angle of refraction by the following equation

$$n = \frac{\sin i}{\sin r} \quad \ldots\ldots [b]$$

n = index of refraction.

In preparing Table 11, the value of P_F was first computed by means of [a] and [b] for the various angles in Table A. Then the values of P_T and P_A were computed for single and double glass by means of Equations 7 to 10 of the text. The values of P_{B1} and P_{B2} were then computed by means of Equations 11 and 12 of the text. The values of P_{B1} and P_{B2} in Table A were then plotted on a large sheet of coordinate paper from which the values in Table 11 were then read.

For the computations in Table A, a value of $n = 1.5$ was used. A value of $\frac{1}{8}$ in. was used for the thickness of a single pane of glass. The value of k was computed by means of Equation 7 using the value of 88 per cent for the energy transmitted through single glass at zero degrees angle of incidence as reported by Shaver.[4] If L and t are in inches, the value of k computed in this way is 0.34. For computing the value of P_F at zero angle of incidence, [a] cannot be used. For zero angle of incidence only, the following form of Fresnel's[5] formula must be used:

$$P_F = \left(\frac{n - 1}{n + 1}\right)^2 \quad \ldots\ldots [c]$$

From an inspection of Fig. 8, it is evident that the ray R_1 reflected from the surface B-B, must in turn be split into two parts again when it is intercepted at the surface A-A. Part of this ray is reflected back into the glass again, while part is transmitted through the boundary A-A. In this way a ray of light is reflected back and forth between the two surfaces of the glass, and the intensity of the ray constantly diminishes as it travels back and forth. Because of this internal multiple reflection, Equations 7 to 10 are really only the first terms of a rapidly converging series of which only the first term is important.

Note: When the product kL is small, the quantity e^{-kL} is difficult to evaluate accurately by means of available tables. The following formula, which is the first term of one of the logarithmic series, may be used under these conditions.

$$e^{-kL} = \frac{2 - kL}{2 + kL}$$

*The Trane Co. Member of Board of Consulting and Contributing Editors.

†See list of symbols on p. 446, July issue.

Part 6. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, May, 1938, pp. 315-318; Part 2 in June, pp. 391-394; Part 3 in July, pp. 445-447; Part 4 in August, pp. 521-523; and Part 5 in September, pp. 573-575.

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[5]Hardy and Perrin, The Principles of Optics, pp. 27-28.

For values of kL smaller than 0.50, the error in using this formula is less than 1 per cent.

Appendix 8—Conduction of Heat Through Single Glass When Its Temperature is Raised by the Internal Absorption of Solar Energy

In this appendix a formula is derived for computing the conduction of heat through glass when internal absorption of solar heat takes place. By means of this formula, the fact will be demonstrated that the conduction of heat is reduced only a little by the small amount of energy absorbed by the glass. Evidently the internal energy absorbed by a single thickness of glass is $P_{A1}I_o$. If the temperature of the glass is lower than the temperature of the outdoor air in contact with it, the flow of heat will take place from the outdoor air to the glass as shown in Fig. 9. Inasmuch as the internal absorption of solar energy by clean glass is small, as shown in Fig. 6, and one face of the glass is in contact with cool room air, it is probable that the temperature of the glass will be lower than the temperature of the outdoor air.

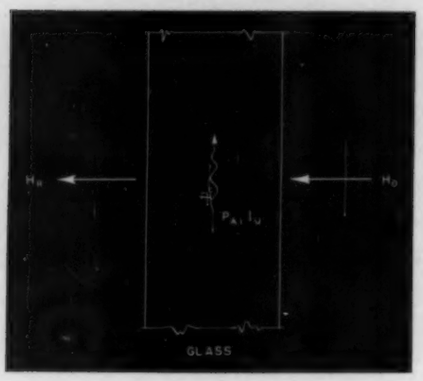


Fig. 9—Conduction and absorption of heat by glass

Referring to Fig. 9, for a heat balance.

$$H_o + P_{A1}I_o = H_R \quad \text{.....[a]}$$

Inasmuch as the conductivity of glass is high and the thickness of ordinary window glass is small, both surfaces of the glass may be assumed to be at the same temperature. Hence,

$$H_o = f_o(t_o - t_g) \quad \text{.....[b]}$$

$$H_R = f_R(t_g - t_R) \quad \text{.....[c]}$$

Eliminating t_g between [b] and [c],

$$\frac{H_R}{f_R} + \frac{H_o}{f_o} = t_o - t_R \quad \text{.....[d]}$$

Eliminating H_o between [a] and [d], and assembling terms,

$$H_R\left[\frac{1}{f_R} + \frac{1}{f_o}\right] = (t_o - t_R) + \frac{P_{A1}}{f_o}I_o \quad \text{.....[e]}$$

For glass, because of its high conductivity and thinness, the following equation is commonly used,

$$\frac{1}{U} = \frac{1}{f_R} + \frac{1}{f_o} \quad \text{.....[f]}$$

Hence, substituting [f] into [e], and solving for H_R,

$$H_R = U(t_o - t_R) + \frac{U}{f_o}P_AI_o \quad \text{.....[g]}$$

The above equation represents the sum of the heat conducted through the glass from the outdoor air and the heat absorbed by the glass. However, as it stands above [g] does not tell the true story. The value of U/f_o in [g] is about 0.55 ($U = 1.1$, and $f_o = 2.0$). What becomes of the other 45 per cent of the radiant energy absorbed internally by the glass? Also it is evident that if the glass temperature is raised slightly by absorbed energy, the conduction heat gain must be less than the value represented by the first term $U(t_o - t_R)$ in [g]. The conduction of heat from the outdoor air to the glass must be reduced if the glass temperature is raised by solar absorption. Hence, to cast [g] into such a form that its terms will present a true picture of the action taking place, rewrite it in the following form,

$$H_R = U(t_o - t_R) + \frac{U}{f_o}P_AI_o + P_AI_o - P_AI_o \quad \text{.....[h]}$$

Table A—Computation of P_{H1} and P_{H2} for Various Angles of Incidence

Angle of Incidence i	cos i	P_r	P_{T1}	P_{A1}	P_{T2}	P_{A2}	P_{H1}	P_{H2}
0	1.0000	0.0600	0.8833	0.0399	0.7802	0.0752	0.9232	0.8155
15	0.9659	0.0599	0.8825	0.0405	0.7788	0.0763	0.9230	0.8146
30	0.8660	0.0584	0.8780	0.0423	0.7703	0.0794	0.9203	0.8081
45	0.7071	0.0498	0.8596	0.0447	0.7389	0.0832	0.9043	0.7774
60	0.5000	0.0108	0.7875	0.0462	0.6202	0.0826	0.8337	0.6566
65	0.4226	0.0790	0.7324	0.0457	0.5364	0.0792	0.7781	0.5699
70	0.3420	0.0289	0.6506	0.0440	0.4234	0.0727	0.6947	0.4520
71	0.3256	0.0156	0.6297	0.0435	0.3965	0.0708	0.6732	0.4239
72	0.3090	0.0008	0.6070	0.0428	0.3684	0.0688	0.6498	0.3944
73	0.2924	0.0845	0.5824	0.0421	0.3392	0.0667	0.6245	0.3637
74	0.2756	0.0666	0.5560	0.0413	0.3091	0.0643	0.5973	0.3321
75	0.2588	0.0469	0.5277	0.0404	0.2785	0.0617	0.5681	0.2998
76	0.2419	0.0250	0.4971	0.0393	0.2471	0.0588	0.5364	0.2666
77	0.2250	0.0011	0.4648	0.0381	0.2160	0.0559	0.5029	0.2337
78	0.2079	0.0745	0.4302	0.0368	0.1851	0.0526	0.4670	0.2009
79	0.1908	0.0448	0.3931	0.0353	0.1545	0.0491	0.4284	0.1684
80	0.1737	0.0122	0.3543	0.0336	0.1255	0.0454	0.3879	0.1374
81	0.1564	0.0762	0.3138	0.0316	0.0985	0.0416	0.3454	0.1084
82	0.1392	0.0360	0.2715	0.0295	0.0737	0.0375	0.3010	0.0817
83	0.1219	0.0917	0.2285	0.0271	0.0522	0.0333	0.2556	0.0584
84	0.1045	0.0421	0.1847	0.0244	0.0341	0.0289	0.2091	0.0386
85	0.0872	0.0873	0.1417	0.0214	0.0201	0.0245	0.1631	0.0231
86	0.0698	0.0261	0.1004	0.0180	0.0101	0.0198	0.1184	0.0119
87	0.0523	0.0580	0.0629	0.0143	0.0040	0.0152	0.0772	0.0049
88	0.0349	0.0184	0.0311	0.0101	0.00097	0.0104	0.0411	0.0013
89	0.0175	0.0062	0.0087	0.0053	0.0000.6	0.0054	0.0141	0.00012
90	0	0	0	0	0	0	0	0

Clearing and assembling terms,

$$H_R = \left[U(t_o - t_R) - \left(1 - \frac{U}{f_o}\right)P_AI_o\right] + P_AI_o \quad \text{..[j]}$$

Equation [j] presents a true picture of the action taking place. The total absorbed energy, P_AI_o, must be transmitted through the glass into the room if the temperature of the glass is lower than the temperature of the outdoor air. Also because the temperature of the glass is raised slightly by the absorbed energy, [j] shows that the normal conduction heat gain, $U(t_o - t_R)$, through the glass is reduced by the quantity $\left[1 - \frac{U}{f_o}\right]P_AI_o$. The value of this last quantity reduces to 0.45 P_AI_o. Inasmuch as the value of P_AI_o is relatively small to begin with, the entire quantity may be neglected in the interests of simplicity. The small error is on the side of safety. Hence, [j] may be written,

$$H_R = U(t_o - t_R) + P_{A1}I_o \quad \text{.....[k]}$$

It is evident from the preceding discussion that the reduction in the conduction heat gain because of the absorbed solar energy is small in a single thickness of glass.

In order to find the total heat gain through the glass, the value of the solar energy transmitted through the glass must be added to [k]. The solar energy, H_S, transmitted through the glass is $P_{T1}I_o$. Hence

$$H_R + H_S = U(t_o - t_R) + P_{A1}I_o + P_{T1}I_o \quad \text{.....[l]}$$
$$= U(t_o - t_R) + (P_{A1} + P_{T1})I_o$$

By means of [11] of the text,

$$H_R + H_S = U(t_o - t_R) + P_{H1}I_o \quad \text{.....[m]}$$

The procedure outlined in the text for computing the solar and conduction heat gains through single glass is based on [m] above. The same procedure is used for double glass. The error involved in doing so is probably small.

Appendix 9—Derivation of Equation 13

$$D_S = a\frac{P}{f_o}I_o - (1 - P)D$$

Refer to Fig. 10. Of the incident solar energy I_o impinging on each square foot of an opaque surface only a fraction aI_o is absorbed by the surface; the balance is reflected back into space. The amount absorbed depends upon the color and texture of the surface. The darker and rougher the surface, the greater is the value of a, the coefficient of absorption.

Of the heat aI_u absorbed by the surface, a portion H_o is conducted through the outside surface film back to the outdoor air, because the temperature of an opaque surface exposed to sunlight is normally higher than the temperature of the outdoor air. Hence, of the heat aI_u, a smaller amount H_w starts the trip into the interior of the wall, where, as is evident from Fig. 10,

$$H_w = aI_u - H_o \quad \text{.....[a]}$$

Now, because the temperature of the interior of the wall is raised by the heat H_w traveling through the wall, only a fraction of H_w reaches the interior surface of the wall and the room air in contact with this interior surface.

Hence, the value of H_R may be said to be related to H_w by

$$H_R = PH_w \quad \text{.....[b]}$$

P = fraction of H_w that reaches the interior of the room. Eliminating H_w between [a] and [b], and clearing,

$$H_R = aPI_u - PH_o \quad \text{.....[c]}$$

Now

$$H_o = f_o(t_{so} - t_o) \quad \text{.....[d]}$$

t_{so} = temperature of outside surface of wall.

Substituting [d] into [c],

$$H_R = aPI_u - Pf_o(t_{so} - t_o) \quad \text{.....[e]}$$

Also the heat flow through the wall is caused by the difference in temperature of the two surfaces of the wall. Hence, approximately,

$$H_w = C(t_{so} - t_{sR}) \quad \text{.....[f]}$$
$$H_R = PC(t_{so} - t_{sR}) \quad \text{.....(g)}$$

C = over-all conductance of the wall alone. (Film coefficients not included.)

t_{sR} = temperature of interior surface of wall.

Also

$$H_R = f_R(t_{sR} - t_R) \quad \text{.....[h]}$$

Eliminating t_{sR} between [g] and [h] and reassembling terms,

$$H_R\left[\frac{1}{PC} + \frac{1}{f_R}\right] = (t_{so} - t_R) \quad \text{.....[j]}$$

$$\frac{H_R}{P}\left[\frac{1}{C} + \frac{P}{f_R}\right] = (t_{so} - t_R) \quad \text{.....[k]}$$

Eliminating t_{so} between [k] and [e], and reassembling terms,

$$\frac{H_R}{P}\left[\frac{1}{f_o} + \frac{1}{C} + \frac{P}{f_R}\right] = (t_o - t_R) + \frac{a}{f_o}I_u \quad \text{.....[l]}$$

Now by the following familiar equation,

$$\frac{1}{U} = \frac{1}{f_o} + \frac{1}{C} + \frac{1}{f_R} \quad \text{.....[m]}$$

However, the quantity in the parentheses on the left side of [l] is different from [m] because the last term is P/f_R instead of $1/f_R$ as in [m]. However, the error will be small if we let

Fig. 10—Diagram illustrating travel of solar heat through walls and roofs

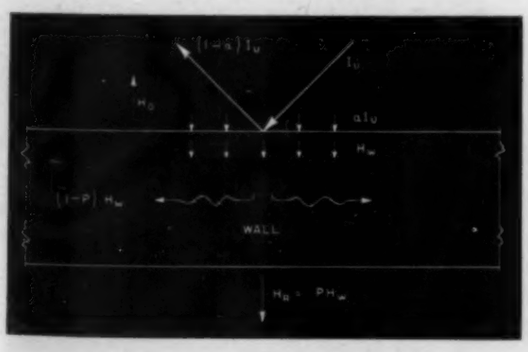


Table B

	Test Panel	Tabulated Date [1]	Time Lag [2]	Time of Maximum Heat Flow [3]	Maximum Heat Flow [4] H_R	Outdoor Temperature [5] t_o	Test Room Temperature t_R	$t_o - t_R = D$	Maximum Heat on Horizontal Surface [6] I_u	Conductance [7] C	Inside Film Coefficient [8] f_R	U	P
1	2 in. Pine	7/27	1 hr. 40′	3–4 p.m.	24.5	97.5	70	27.5	185	0.83	1.9	0.45	0.49
2	6 in. Concrete	9/10	2 hr. 40′	3–4 p.m.	28.5	97.9	70	27.9	125	8.00	1.9	0.87	0.39
3	4 in. Gypsum	9/10	2 hr. 25′	3–4 p.m.	17.0	97.9	70	27.9	125	1.445	1.9	0.58	0.35
4	4 in. Cork-Conc.	9/10	3 hr. 0′	3–4 p.m.	11.5	97.9	70	27.9	125	1.218	1.9	0.54	0.25
5	4 in. Conc.-Cork	7/27	2 hr. 0′	3–4 p.m.	17.5	97.5	70	27.5	185	1.218	1.9	0.54	0.29

The page numbers below refer to the 1932 A.S.H.V.E. paper of Footnote C.
[1] From Table 4 on p. 247.
[2] From Table 3 on p. 246.
[3] From Table 4 on p. 247.
[4] From Table 4 on p. 247.
[5] From Table 3 on p. 246.
[6] Read from curves on pp. 250 and 251.
[7] From Table 1 on p. 236. In this table, these values are headed k.
[8] From Table 1 on p. 236. In this table, these values are headed hi.

$$\frac{1}{U} = \frac{1}{f_o} + \frac{1}{C} + \frac{P}{f_R} \quad \text{.....[n]}$$

Substituting [n] into [l], clearing, and assembling

$$H_R = aU\frac{P}{f_o}I_u - t_o + PU(t_o - t_R) \quad \text{.....[p]}$$

H_R as given by [p] is the total solar heat conducted through the wall. Evidently the amount by which the solar heat gain exceeds the conduction heat gain is $[H_R - U(t_o - t_R)]$, or

$$H_R - U(t_o - t_R) = aU\frac{P}{f_o}I_u - t_o + PU(t_o - t_R) - U(t_o - t_R)$$

$$= aU\frac{P}{f_o}I_u - U(1-P)(t_o - t_R) \quad \text{.....[q]}$$

In order to find the fictitious temperature difference which when multiplied by U will yield $[H_R - U(t_o - t_R)]$, let

$$UD_s = H_R - U(t_o - t_R) \quad \text{.....[r]}$$

Therefore,

$$UD_s = aU\frac{P}{f_o}I_u - U(1-P)(t_o - t_R)$$

$$D_s = a\frac{P}{f_o}I_u - (1-P)(t_o - t_R) \quad \text{.....[s]}$$

As $D = t_o - t_R$

$$D_s = a\frac{P}{f_o}I_u - (1-P)D \quad \text{.....[t]}$$

This is Equation 13 of the text.

Before this equation can be used, it is necessary to find some probable value of P, the percentage of energy transmitted through the wall. Accordingly, the few computations listed in Table B were made from the data in an A.S.H.V.E. paper[c] on the solar heat gain through various roof panels that were tested. From the five tests analyzed, the values in the last column show that the values of P range from 0.25 to 0.49. No claim is made for great accuracy for these results, as too few tests were analyzed, and also the analysis is based on the peak heat transmission of the day. However, considering the approximate character of the estimates for solar heat gains, it is probable that a value of $P = 0.5$ will be satisfactory for ordinary construction. For heavy construction, or well insulated construction, a value of about $P = 0.35$ would probably be satisfactory, to judge from Table B. The values of P in this table were obtained by substituting the tabulated test values into Equation 13 of the text. Inasmuch as all the test panels were black, a value of $a = 0.9$ was used. A value of $f_o = 2.0$ was used as was done in the A.S.H.V.E. paper.[c] As can be seen by referring to Fig. 1 on p. 93 of the 1938 *Heating, Ventilating, Air Conditioning Guide*, the value of f_o for brick, stucco, and plaster surfaces is about 2.0 in still air.

[Concluded]

[c] Houghten, Blackshaw, Pugh, and McDermott; Heat Transmission as Influenced by Heat Capacity and Solar Radiation; *Transactions A.S.H.V.E.*, 1932, p. 231.

Savings Within Six Years Will Repay Money Spent for Modernizing Heating

By Harry Duvendack*

DURING the summer of 1937, the heating and power plant of the National Republic Building, Chicago, was thoroughly modernized. The money spent was quite substantial, but the first year's savings indicate that the investment will be amortized within six years. This is considered quite satisfactory in view of the fact that much of the expenditure was obligatory.

The three 250 hp longitudinal drum water tube boilers had been in service since the building was erected 30 years ago. Proper operation and adequate maintenance had kept these boilers in good condition for their age, but they had reached the point where they had to be either replaced or thoroughly rehabilitated. Cost of rehabilitation versus new boilers was compared with anticipated useful life, and the decision was in favor of rehabilitation.

The elements of the boilers which were in sound condition were retained. New boiler tubes and superheater tubes were put in. The nipples in the headers were replaced. The old non-return valves were taken out, and new valves of more modern type were installed. New brick work was put in throughout. The old Dutch oven type stationary grate stokers were removed from all three boilers. New stokers were installed under two of the boilers, and the third was equipped for gas firing. Additional combustion space was provided by lowering the floors of the coal burning boilers two feet, and the floor of the gas burning boiler three feet. The object in equipping one boiler for gas burning was to take advantage of the low commercial gas rate which prevails during the summer months, and which was inaugurated recently.

Three gas burners and two booster motors were installed on the gas fired boiler. The burners are built on the multi-unit, multi-jet principle; each consists of a number of mixing tubes, each of which is supplied with gas through four jets which meet at a central, predetermined point in the tube, to give an intimate mixture of air and gas which results in thorough combustion and high furnace temperature.

The new stokers are of the single retort underfeed type, with plunger feed and dumping grates. Each has a grate area of 45 sq ft and a maximum coal burning capacity of 1600 lb per hr, which will operate the boilers up to 150 per cent of rating.

Before the modernization, there was no constant knowledge of steam output and steam costs. These factors were known only when tests were made with instruments brought in from outside. Coal handling methods were primitive. Both these deficiencies were remedied.

Coal is now dumped from trucks in the alley into bunkers with 200 tons capacity. The modernization program included a new automatic conveyor which carries coal from the bunkers to the 20 ton overhead hoppers. A new traveling weigh larry carries the coal from the overhead hoppers to the stoker hoppers, weighing all coal as fired and providing an accurate and constant knowledge of coal consumption.

Other new equipment included a steam flow meter and a water flow meter so that one may be checked against the other, a CO_2 meter, a feedwater regulator, soot blowers, a smoke indicator which shows at the boiler front a continuous picture of conditions in the stack, a damper regulator which controls the draft on the coal fired boilers and the rate of coal feed according to the boiler pressure, and electric starting boxes for stokers, house pumps, and the booster motors on the gas lines.

All power and pumping equipment was rebuilt. The pumps—three feed, one house and two vacuum—were thoroughly renovated. New stainless steel rods and new pistons were put in, and the cylinders were rebored. A new 25 hp electric power pump with a capacity of 120 gpm was installed. The three air compressors used for sewage ejection were completely overhauled. The three steam engines were fully reconditioned. These consist of two corliss type single engines, each direct connected with a 150 kw generator, and a four valve engine direct connected with a 100 kw generator.

Beginning August 19, 1938, all power and light for the building have been generated in this plant. A two month test, previously conducted, demonstrated that this could be done with a saving as compared with the cost of purchased current. Substantial power savings in the future are indicated, owing to lower steam costs made possible by the modernization of the boiler plant and by the increased efficiency of the reconditioned engines and generators.

The piping was thoroughly inspected during the modernization. Many test holes were drilled. All piping was found to be in first class condition, with the exception of the blowoffs, which were renewed. The steam traps, which had been put in with the boilers, were found to be in good shape, as the elements had been replaced where necessary.

The heating job in this building of 17 floors and four basements is 36,000 sq ft of radiation. Steam is also used for power, as stated above, for hot water heating, and for operating the pumps and air compressors. An interesting sidelight on the heating system operation is that the building vacuum cleaning equipment is utilized to maintain a vacuum on the heating lines. Because of the age of the piping, it was impossible to get more than a 2 in. vacuum, so a connection was made to the steam engine driven vacuum cleaning equipment, which makes it possible to maintain an average 14 in. vacuum for the heating system.

Steam is delivered by the boilers at 165 lb pressure and about 70 deg superheat. The per cent of rating at which the boilers operate depends on current needs. In winter the coal fired boilers, when two are used,

*Chief Engineer, National Republic Bldg.

operate just below 100 per cent. The gas fired boiler runs as high as 175 per cent of rating.

The boiler feed water is heated by exhaust steam to a temperature of about 200 F. Exhaust steam is also used for heating the building and for providing hot water. It is delivered to the heating lines at 1 lb pressure. The supply of exhaust steam is adequate for all these purposes except when the outside temperature drops below 5 F.

On the new schedule, gas is used for fuel during the seven months of the year (April 1 to October 31) when the low rate is in effect. Coal is burned during the remaining five months, when the cost of gas would be prohibitive. Gas has, in this plant, during the low rate season, proved economical. For instance, in February, 1938, 4,370,000 lb of steam were generated with coal at a cost of $1363.50. The cost per 1000 lb of steam was 31.2 cents. In April, 1938, 4,495,000 lb of steam were generated with gas, using 51,622 therms, at a cost of $1138.95. The cost per 1000 lb of steam was 25.3 cents. The gas fired boiler operates at an efficiency estimated at 74 per cent and the coal fired boiler at about 70 per cent. Gas is most economical, as compared with coal, when operating at high ratings. The average CO_2 with the coal fired boilers is about 13 per cent and with the gas fired boiler about 9 per cent.

During the coal burning period in winter, improved performance and substantial economies have been achieved by the modernization as is demonstrated by the following figures:

WITH OLD EQUIPMENT (1936-37)

	Coal burned, lb	Water evaporated, lb
Nov.	595,900	4,710,000
Dec.	654,300	5,098,000
Jan.	730,440	5,337,000
Feb.	658,000	4,885,000
March	724,200	5,337,000
Total	3,362,840	25,367,000*

*(7.5 lb per lb coal)

WITH NEW EQUIPMENT (1937-38)

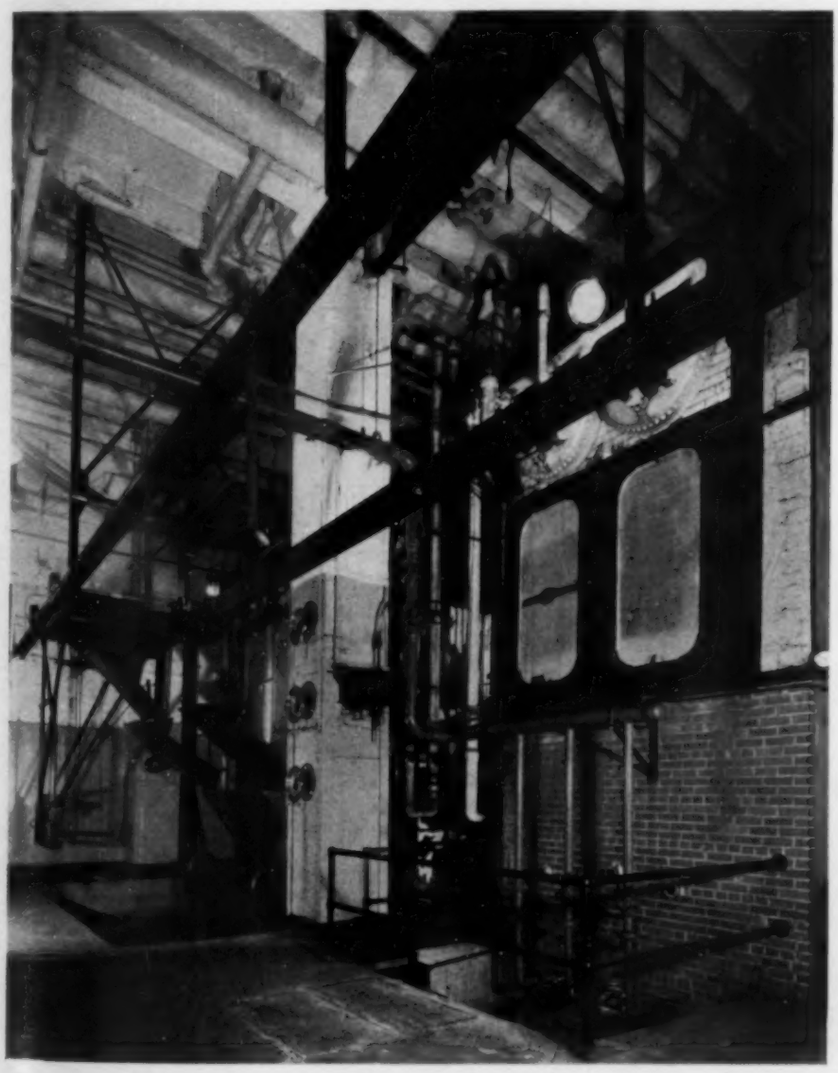
Nov.	410,800	3,347,000
Dec.	606,200	5,040,400
Jan.	569,100	4,900,000
Feb.	540,000	4,370,000
March	538,300	5,494,300
Total	2,664,400	23,151,700†

†(8.7 lb per lb coal)

The results of this modernization work have abundantly justified its cost. This is especially gratifying, considering that a large part of these expenditures would have had to be made regardless of potential economies.

It paid to rehabilitate the building's entire mechanical equipment at the same time. Performance to date indicates that savings will pay for the entire modernization program in about six years, and the economies will continue for long after that.

Maintenance costs on all equipment will be substantially reduced for years to come. On the old equipment, maintenance had been getting more and more costly every year. There should now be little or no maintenance on the boilers, stokers and their auxiliaries during the next five years. This does not mean that maintenance will be neglected; it will be kept at a minimum by frequent inspection and by those timely small repairs which obviate costly future repairs.



Last year the 30-year old heating plant of the National Republic Building, Chicago, was thoroughly modernized and the first year's savings indicate the investment will be amortized within six years. . . . . The three boilers, which have been in service since the building was erected, were rehabilitated, two of them were equipped with modern stokers, and the third with gas burners to take advantage of the low gas rate prevailing in summer months. . . . . In this view of the boiler room, the gas fired boiler is at the right

Van stoning copper tubing is a typical oxy-acetylene heating application

Oxy-Acetylene Heating for Bending, Straightening and Forming

By H. H. Moss*

THERE are hundreds of forging, forming, straightening, bending and pressing operations that can be performed with the aid of the intense, localized heat of the oxy-acetylene flame. Of late, the use of the oxy-acetylene flame for work of this nature has grown and has been facilitated by the availability of large capacity heating blowpipes, which have been introduced during the past few years, and the even more recent multi-flame heating heads.

The oxy-acetylene flame has many advantages as a source of heat. One of the most important is the speed with which heating operations can be performed. With temperatures approaching 6000 F available instantly, a very steep temperature gradient can be established between the source of heat and the work, and as the heat is highly concentrated, it flows rapidly into a very small area surrounding the flame. The result is a rapid elevation of the temperature of the localized area of the work. Temperatures suitable for forming or bending operations can thus be obtained in a matter of moments.

Portability is another important advantage. The oxy-acetylene blowpipe saves time by making it unnecessary to dismantle a large structure or machine in order to heat the particular part to be bent or straightened. By means of this process, the source of heat is readily carried to the work, rather than the work to the source of heat as is usually necessary with other methods.

Still another important and valuable feature of localized oxy-acetylene heating is the ability of the operator to regulate at will the heat intensity across a section. This enables controlled forming of a section which can be accomplished either by stretching the outside fibers of the section or by compressing the inside fibers, the latter method usually being preferred.

Finally, although oxygen and acetylene would seem, on first thought, to be a more expensive fuel than coke,

*Engineer, Linde Air Products Co. Condensed from an article in *Oxy-Acetylene Tips.*

coal, city gas, oil, or gasoline, the intensity of the oxy-acetylene flame and the ease with which it can be directed make it possible to do the work with considerably less fuel than when other processes are used. Actually, the fuel cost with the oxy-acetylene flame is less, in most cases, than with other fuels. With other sources of heat, it is frequently necessary to heat the entire part in a furnace in order to permit a forming operation at one point, or along the edge, as in forming a flanged rim. With the oxy-acetylene flame, heat is applied only where needed, and thus less fuel is required. Also, there is a marked saving of time and labor.

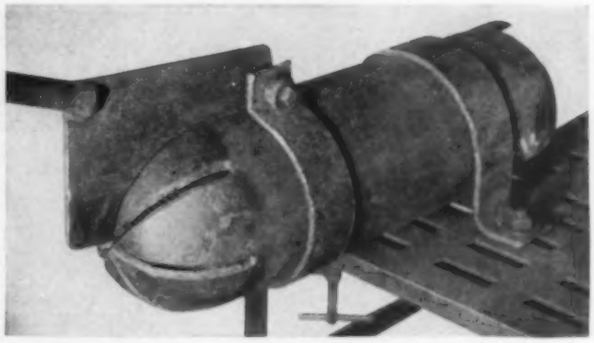
Typical Applications

The fabrication of pipe fittings in both shop and field affords an example of the application of "oxy-acetylene smithing" in conjunction with oxy-acetylene cutting and welding. Bends, semi-welded turns, welded reducers, and orange peel and formed heads are readily made in this way. In the copper shop, flanges in copper pipe are turned up quickly and at small expense with heat from the blowpipe.

One of the more recent developments in the overland piping field uses oxy-acetylene smithing on a rather extensive scale in the production of wrinkle bends in place of the more expensive and tedious fire bends.

In boiler work, typical applications include laying up plate edges, boiler head flanges, and lapped plates. When a flange or head is to be welded on a large tank which is slightly out of round, heat from the oxy-acetylene flame permits the defect to be corrected readily by hammering.

Ironing tanks and process equipment by oxy-acetylene smithing is a typical application encountered in the average fabrication shop. A great deal of this work is done on the vessel or tank during the course of general fabrication in order to insure neat and snug fits which are



Orange peel heads are easily fabricated by means of this simple jig and heat from the blowpipe

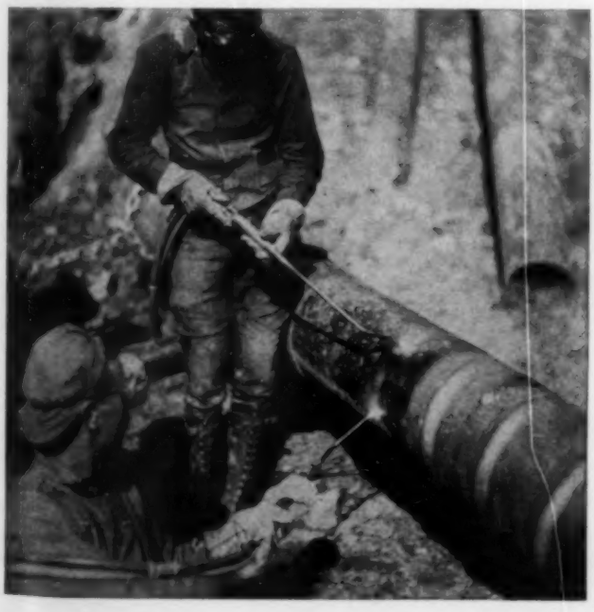
difficult and more costly to obtain by ordinary smithing methods. The operation proceeds rapidly, thereby eliminating much idle time which otherwise would be consumed in scheduling the job and waiting on the forge.

Another procedure which will doubtless find many applications in industry utilizes localized oxy-acetylene heat to adjust the length of wire, bars, piping and similar materials by means of local upsetting or stretching. With the aid of a press or jacks, or other suitable force, a section of the object can be readily shortened or lengthened to minute degrees of adjustment.

In addition, there are many other straightening jobs that should not be overlooked. Sheet metal buckets, ladles, tanks, kettles, pots and stills, all of which are constantly being dented or bent, provide many opportunities for straightening work.

The selection of the most efficient unit for any heating job will depend on the nature of the job. Where the area to be heated is small, the single flame blowpipe can be used to advantage. On the other hand, the multi-flame heads are usually more efficient than single flame heads where the area to be heated is not confined to a single spot.

Heavy duty heating blowpipes facilitate the wrinkle bending of overland pipe lines



Centrifugal Water Vapor Compressor

As there are many who are not familiar with the centrifugal water vapor refrigeration compressor used in air conditioning, a brief explanation of the principle is probably in order. At different pressures water has different boiling points. For example, the boiling point at 100 lb absolute pressure is 327.8 F. At atmospheric pressure of 14.7 lb (*i. e.*, in an open vessel) the boiling point is 212 F and each pound of water turned into steam and taken from the vessel in this form carries away with it over 1000 Btu. In the same way, water at a pressure of 0.2 lb per sq in. absolute (spoken of as 29.6 in. of mercury, vacuum) boils at 54 F, and each pound of steam so taken off carries a substantial quantity of heat with it. The cooling effect is therefore produced—paradoxical as it may seem—by boiling water at low temperatures. There is no chemical refrigerant.

The refrigerating unit consists of five major parts—an evaporator chamber, which partly comprises the base of the unit, the motor driven centrifugal compressor, a vacuum pump, a chilled water pump, and a condenser.

Thermal Expansion Valves

By W. A. Stains*

POSSIBLY one of the most essential parts of a cooling system is the thermal expansion valve. Much could be written on this one piece of equipment, but the things it doesn't do are our particular problems here.

A little study of the duties of a thermal expansion valve may be in order. Most engineers know that a thermal bulb inserted into or strapped on a suction line opens and closes a valve but few realize that here is a control device that actually is endeavoring to control something that has already happened—an upstream control, as it were.

The functioning of the thermal valve depends on temperature and pressure—the temperature felt by the remote bulb and the pressure in the evaporator. The temperature felt by the bulb is resolved into a pressure corresponding to the temperature-pressure properties of the particular gas or liquid used in the bulb. This pressure is exerted on one side of a diaphragm or bellows while the evaporator pressure is exerted on the other side of the bellows or diaphragm (see illustration). For the sake of brevity we will use the term "diaphragm" as applying to both itself and the bellows. If the two pressures are equal over the same area there is no movement of the diaphragm in either direction. However, if the pressure on one side increases or decreases, the diaphragm moves toward the lower pressure side. The actuating force necessary to move the diaphragm may come either from the thermal or feeler bulb or from the pressure variations in the evaporator.

If the valve pin is forced away from the seat by the movement of the diaphragm toward the evaporator side and the pin returns to its seat by the opposite movement we can understand that four things may cause the valve to open or close. First, with any given pressure (which will be termed unit pressure) on the bulb side of, or "over," the diaphragm and less than unit pressure on the evaporator or "under" side of the diaphragm the pin will leave the seat of the valve and open. Second, if unit pressure exists under the diaphragm and more than unit pressure exists over the diaphragm the pin will leave the seat and open. If these two conditions are reversed and the pressure over the diaphragm is less than unit, with unit pressure under the diaphragm the pin will press harder against the seat or move toward the seat and close, and the same thing happens when there is unit pressure over the diaphragm and unit plus under the diaphragm. It is evident therefore that any variation of conditions tending to unbalance the two pressures will affect the position of the valve with respect to the seat.

*Sales Engineer, Alco Valve Co., Inc.

If a "Freon" evaporator is assumed to have no pressure drop through it and the suction pressure is 37 lb per sq in. (which corresponds to a saturation temperature of 40 F) the pressure under the diaphragm is 37 lb. If the refrigerant is not being evaporated entirely in the evaporator and reaches the position in the suction line where the feeler bulb is situated it is evident the temperature is 40 F at the bulb. It is further assumed there is no lag between the temperature of the refrigerant and the liquid or gas in the feeler bulb. Since the liquid or saturated gas in the power assembly of the valve is also at 40 F there must be a pressure in the power assembly of 37 lb and there is equal pressure on both sides of the diaphragm and no movement takes place.

Effect of Spring

Since all thermal valves are not quite as simple in construction as this, there is a spring introduced to create a force under the diaphragm and hold the valve shut; to open the valve, this spring force, as well as the force of the evaporator pressure, must be overcome. If the "no pressure drop" evaporator is still used with this valve having a spring in it, it is evident that at 37 lb above the diaphragm and 37 lb plus a spring force of 5 lb the force of 42 lb must overcome the 37 lb and the valve will be closed. (For simplicity, the diaphragm area is assumed as 1 sq in.) But if the valve remains closed no more refrigerant will pass through the valve. Two things can and will happen. One is that the compressor will pump down the suction pressure and if nothing else happens, eventually the evaporator pressure will drop below 32 lb when the force above the diaphragm will begin to overcome the 5 lb spring and the 32 lb force of the evaporator pressure, and at 31 lb evaporator pressure the pin will move off the seat and admit more "Freon" to the evaporator. This cycle would tend to balance and the valve would function as a constant pressure expansion valve only.

However, the other thing that happens is that as the valve closes, the refrigerant in the evaporator is evaporating. The line of complete evaporation in the coil recedes from the point where the bulb is located and the saturated gas becomes superheated because it still picks up a few more Btu's; the farther the refrigerant saturation point recedes from the bulb location the more the gas is heated above the saturation point. Each degree increase in temperature of the gas is another degree of superheat.

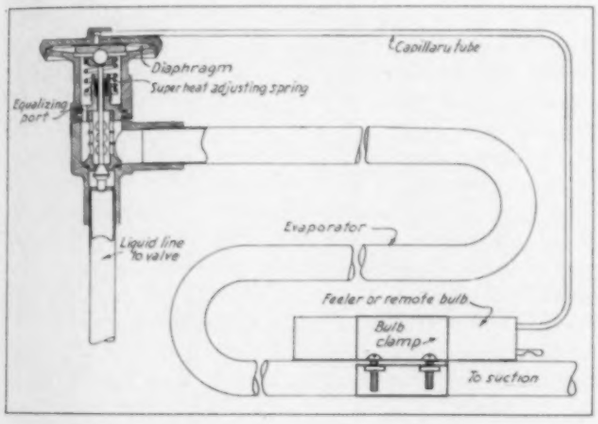
So two things are happening. One is that the pressure is dropping and the gas is increasing in temperature. The result is that the drop in pressure reduces the force under the diaphragm while at the same time the gas temperature is increasing the temperature of the feeler bulb and this in turn is increasing the pressure over the diaphragm in accordance with the pressure of saturated gas in the power assembly. The valve then

opens before the pressure drops materially in the evaporator, admitting more refrigerant to keep up the pressure and reduce the superheat. The spring therefore functions as a superheat regulator in the valve and as the force of the spring is increased or decreased the actuating superheat of the gas must increase or decrease, as the case may be.

The properties of saturated "Freon" vapor in the air conditioning range of approximately 35 to 45 deg are such that for each degree increase or decrease in the saturation temperature the corresponding pressure increases or decreases approximately 1 lb. Therefore, if the valve has a 5 lb force due to spring tension and friction of operating parts it will require practically 5 deg superheat of the suction gas to open the valve. Some valves have an operating gradient for full stroke as low as 3 lb while others have as high as 7 lb.

The assumption above was based on a "no pressure drop" evaporator. Manifestly there is no such thing, for all evaporators have some pressure drop across them. Hence a coil with 10 lb pressure drop presents still another angle to the operation of the thermal valve.

If the evaporator pressure is to provide an average 40 deg saturation temperature of refrigerant in the coil, at the inlet of the coil the pressure is approximately 42 lb and the outlet approximately 32 lb. If the under



Sectional view showing hook-up of diaphragm type thermal valve

diaphragm pressure is secured from the inlet of the coil as in the case of an internally equalized valve the pressure would be 42 lb.

If the valve were set for 10 deg superheat operation the force above the diaphragm would have to overcome the 42 lb pressure plus the 10 deg superheat setting which would require a force above the diaphragm of approximately 52 lb which corresponds to a saturation temperature of about 55 deg. Since the feeler bulb would be affected by the temperature at the outlet of the coil which is at 32 lb suction pressure (which corresponds to a saturation temperature of almost 35 deg) it is seen that a superheat of 20 deg or more would be required to operate this valve and apparently so great a portion of the coil would be given over to providing superheated gas to open the valve that the operation would be beneath full coil capacity. This is a common fault found in the application of thermal valves. Apparently the thermal valve is not of sufficient capacity to supply the coil with refrigerant but actually the valve is doing all it can. A change from internal to external equalizer valves will correct a condition of this kind.

Type and Location of Bulb

The proper application of remote bulbs is essential, as is the style of bulb used. Insert type bulbs are better suited to close control of comparatively short pass evaporators or when low superheat is desirable and quick response of the valve is required. It is desirable to install insert type bulbs in a tee of the suction line not affected by other valves in a group and also at a location which has one or two short pattern elbows between it and the suction outlet from the coil. All feeler bulbs should be kept away from places where liquid or oil might be trapped about the bulb. If the bulbs are in trapped locations or if they are installed where impingement of oil and refrigerant particles or droplets might occur, insufficient refrigerant will be fed to the coils and low suction pressures will be evidenced as well as high air temperatures and general dissatisfaction. (The term "impingement" as used above refers to the droplets and particles of oil and liquid refrigerant that are carried along in the suction gas stream and adhere or impinge upon the bulb).

Thermal valves should be selected for the capacity of the evaporator on which they will operate. Oversizing of valves is detrimental to good control and if oversizing is calculated to remove some objectionable feature such as gas in the liquid line, it is poor economy and poor engineering. Indifferent results will be secured at best.

High velocity distributors, dependent upon the velocity of the refrigerant, require close coupling of the valve outlet to provide the velocity by the jet action of the valve. Weir type distributors should ordinarily be provided with some means to break down the jet velocity before entering the header. This might be done by installing elbows or return bends to secure the desired results or long connections of large pipe between valve outlet and header may be used.

Very often the location of the remote bulb may be changed to promote better results in the coil operation. Where several coils with individual valves are used on the coils or more than one valve on a coil the remote bulb should be on the suction line serving that coil or portion of coil independently of other suction lines.

Cases have been observed where two valves feed into two sections of a coil and a common suction header serves both sections. The two remote or feeler bulbs are attached to the common suction header or line. Obviously, the individual valves cannot be controlled from the common point as the conditions in each part of the coil will not be the same. It is highly essential that each valve be controlled from a bulb on the suction connection serving only that section of coil.

Attempts have been made to control the valve from a point on one of the tube passes or circuits where it enters the common suction header but only in case of equal loading on all the circuits can anything even ap-

proaching good control be expected; such cases are so few and far between that this method should be avoided.

Often a thermal valve indicates insufficient capacity due to its application. A simple method to determine if a valve passes sufficient liquid to maintain desired suction pressures is to remove the bulb from the suction line and hold it in the hand or leave it suspended in the warm air. If this fails to bring up the pressure, before condemning the valve check for line stoppage, gas in the liquid line (which usually is accompanied by a hissing or whistling noise in the valve), air loadings and other sources of trouble outside the valve.

Sometimes the thermal valve apparently does not respond to adjustment. Regardless of this situation, it is probable that the valve is responding to adjustment but some other condition exists that offsets any adjustment that can be made. One of the first things to look for is the location of the thermal or feeler bulb. Cases have been found where the valve apparently couldn't be adjusted to maintain a superheat condition of the gas lower than 20 or 25 deg. It was observed the erecting engineers had placed the bulb at a point where impingement was occurring and either the valve was closed off and produced a high degree of superheat at the bulb, or there was complete saturation at the bulb.

This is explained by the fact that the bulb was directly in line with a tube circuit outlet and whenever liquid was fed to the coil it traveled down to the last pass or around the last return bend in the circuit where the velocity of the gases carried liquid refrigerant straight through that last straight run and directly onto the spot in the header where the remote bulb was located. The trouble was easily corrected by simply moving the bulb to a location where such impingement didn't occur.

The adjustment of thermal valves can be quite satisfactorily made by the use of a simple dial thermometer having a remote bulb that can be fastened to the suction line near or at the thermal or feeler bulb of the valve. Knowing the suction pressure at the outlet of the coil which is reflected on the suction gage at the compressor and estimating the drop from coil to compressor, then adding this drop to the gage, the pressure at the coil outlet can be determined. Then read the corresponding saturation temperature of that pressure from a table of the properties of refrigerant, increase it by the desired superheat and open or close the valve until the temperature on the dial thermometer corresponds approximately to the figure arrived at. Quite likely the valve might be opened to operate at a lower superheat setting than was originally intended and if this can be done without raising the suction pressure too far for the machine and motor to handle, it will increase the capacity of the coil by giving a greater heat transfer to the coil.

Thermal valves, through defective manufacture, careless handling or damage will lose their charge. When this occurs the valve will remain closed or practically so. This can be and is easily determined by most service engineers. However, if the valve loses only part of its charge it may function to some extent but will not open wide enough to supply the proper quantity of refrigerant. Also, through inefficient manufacturing methods or carelessness, the thermal or power element may be incorrectly charged and this will show up like a valve which has partially lost its charge. The cure for bad power elements or valves is replacement with good ones.

High suction pressures are attributed to an unbalanced condition in compressor and evaporator where the evaporator is of too great a capacity. If the quantity of liquid refrigerant evaporated is too small, the suction pressure is low. Inversely, if the refrigerant is evaporated too fast for the compressor to pump out that quantity of gas, the suction pressure rises. It may be the evaporator is too large for the compressor capacity and this may be remedied by speeding up the compressor if there is available power at hand and the compressor isn't rotating at its highest allowable limit. If so, and the conditions of air temperatures can be obtained with less evaporator, the thermal valve may be set with a higher superheat adjustment which will reduce the effective capacity of the coil.

If the overload is determined as the oversupply of air volume or higher temperature air, it is obvious that the air volume or temperatures must be reduced.

Sometimes valves are selected with too great a capacity or the liquid head pressures are too high. Oversized valves will cause high suction pressures and this is usually accompanied by cold crank cases and even cold discharge from the compressor. The same conditions may exist from high liquid pressure at the valve.

The result of the oversized valve is to feed the evaporator spasmodically. This may allow the refrigerant to flow into the tubes in greater quantity than can be evaporated before liquid reaches the thermal bulb location. If this happens, liquid refrigerant may pass on into the suction line and even back to the compressor. In the meantime, the bulb "feeling" the saturation of the refrigerant gas closes off the valve. If the evaporator is a long pass type (or in other words, the tube circuits are unusually long) there may not be any refrigerant entering the evaporator momentarily. There being more liquid refrigerant in the tube than can be evaporated and with this liquid refrigerant working its way to the outlet end of the coil, the gas leaving the tube is saturated. However, the liquid that adheres to the tube walls at the entrance section of the circuit will have been evaporated and the gas thus formed following the liquid can be conceived as being superheated slightly and becomes more so as it passes through the tube, so there will be superheated gas following the liquid past the thermal bulb. Until all the liquid has passed the bulb, the valve will remain closed, but the instant the gas which formed at the entrance of the circuit arrives at the bulb location, the valve will open wide and again feed more refrigerant than the tube circuit can evaporate. This cycling continues indefinitely, and the hunting condition is more exaggerated the more oversized the valve is.

This cycling liquid feed may even show a decided hunting in the suction pressure gage and at one point in the cycle the suction line may show frost at low back pressures and sweating or dryness at the other extreme with high suction pressures.

Insensitive thermal valves may also show a condition similar to this just described but by virtue of its insensitivity the hunt may not be so violent—neither will the overall operation of the system be as good. The pressure fluctuations won't be as sharp and both flooding through and pumping down will show longer cycles.

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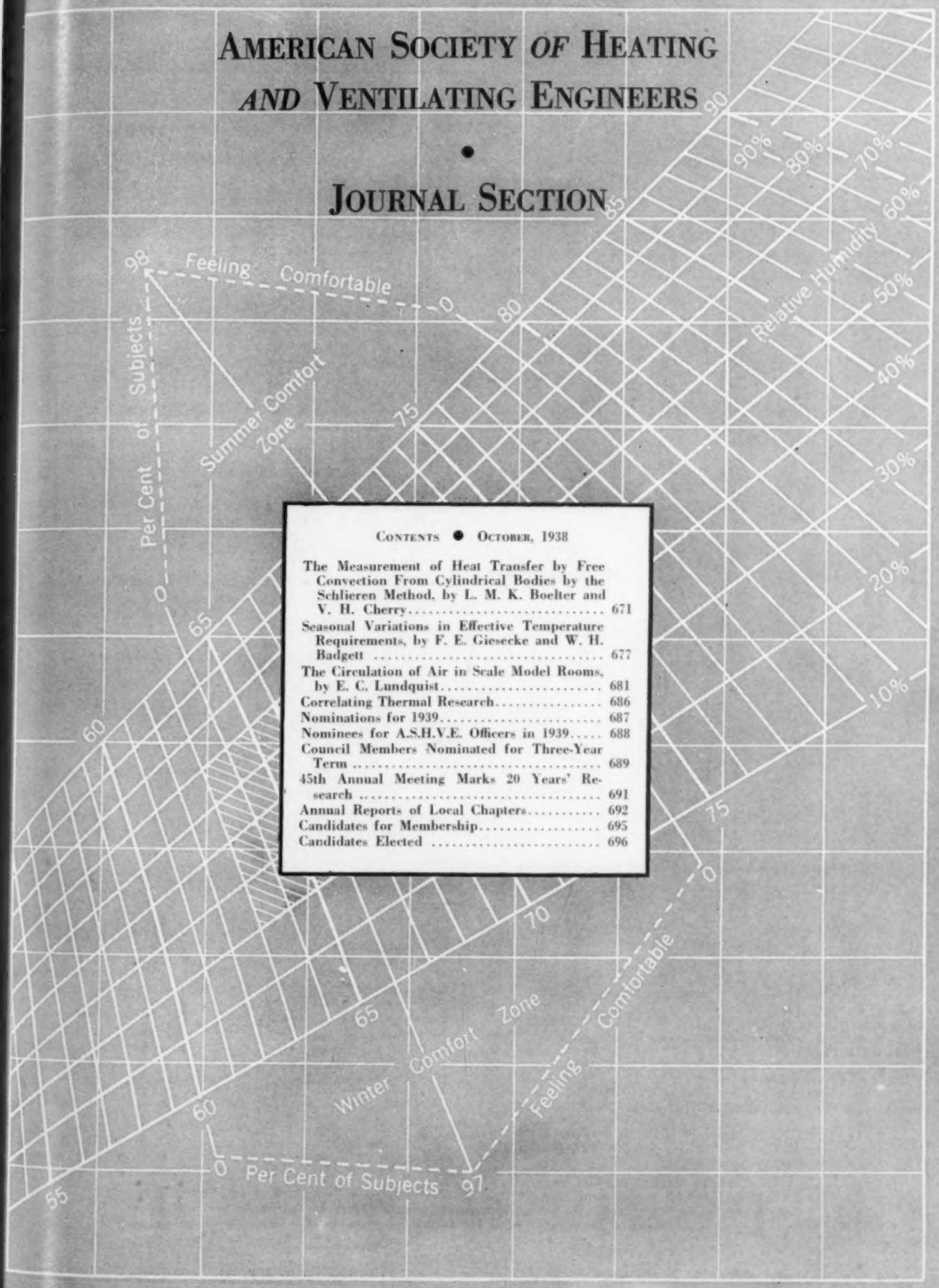

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SOCIETY MEETINGS

1939

45th Annual Meeting

Wm. Penn Hotel, Pittsburgh, Pa., January 23-26, R. A. Miller, General Chairman

1940

46th Annual Meeting

Hotel Statler, Cleveland, Ohio, January 22-26, L. T. Avery, General Chairman

Sixth International Heating and Ventilating Exposition, Public Hall

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The Measurement of Heat Transfer by Free Convection From Cylindrical Bodies by the Schlieren Method

By L. M. K. Boelter* and V. H. Cherry† (*MEMBERS*), Berkeley, Calif.

THE rate of heat transfer from the surface of a body surrounded by a diathermanous fluid is due to convection and radiation to the surroundings. Often in experimental technique, the free convection rate is calculated as a difference between the heat rate input and the rate of radiant energy transfer. If the latter term is not measured separately, the use of an estimated emissivity may introduce a large error.

There are several means by which the error in the determination of heat transfer by free convection due to the uncertainties of the emissive power of a surface may be reduced or eliminated.

1. The rate of heat transfer from solids to liquids by convection is so large that the radiation is negligible. By the correlation with generalized variables (dimensionless moduli) the data determined in this manner may be extrapolated to other fluids.

2. The radiation from small wires may be negligible in comparison with the convective heat losses.

3. The solid may be given a highly polished surface so that an error of 50 per cent in the emissive power may amount to an error of only 5 per cent in the determined convective losses.

4. By means of the Schlieren method developed by Schmidt,[1] the temperature gradient in the fluid near the surface of the solid may be determined. The temperature gradient multiplied by the thermal conductivity of the fluid results in the convective heat rate per unit area. The disadvantages of the method are:

a. The thermal conductivities of fluids are not accurately known. Variations of 10 per cent in published data may be noted.[2]

b. The method is only applicable to surfaces of such shape as may be created by a straight generatrix along which the temperature is uniform. The stagnant film of fluid on the surface must be between 0.02 in. and 0.10 in. in thickness.

c. These disadvantages are offset by the supplementary data which may be obtained. In addition to the total heat rate by convection, the film conductance as a function of distance along the directrix of a cylinder may be determined. This is an aid, for example, in computing the thickness of ice and frost on the expansion coils of a refrigeration system.[3] The resulting photographic data are also an aid in the visualization of the free convection process.[4]

Data obtained by means of the Schlieren method are employed in this paper to determine the effect of some simply shaped surroundings upon the convective heat rate from a horizontal circular cylinder.

In order to generate confidence in the Schlieren experimental technique, the measured film conductances by free convection in air from a horizontal cylinder far from surroundings are compared with data from the literature.

Experimental Technique

The rate of convective heat transmission may be shown to depend upon the slope of a beam of light as it leaves the thermal field near surface.[5] The light beam is so directed as to be tangent to the surface at the entrance to the thermal field. The beam slope is determined experimentally.

Fig. 1 is a schematic sketch of the apparatus arrangement which was used to determine the heat exchange rate from a horizontal cylinder to air.

An arc lamp was used as a source of light. To insure that all of the light beams were parallel to the surface of the cylinder, the cylinder was slightly tapered.

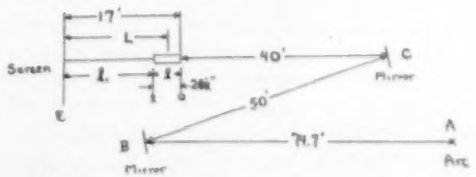

Fig. 1—Schematic sketch of apparatus to determine heat exchange rate from a horizontal cylinder to air

To reduce the taper, and so that the arc lamp might be considered substantially a point source it was necessary that the cylinder be located at a considerable distance from the lamp. This result was accomplished by reflecting the light from plane mirrors as shown in Fig. 1. The distance from the lamp to the *O* end of the cylinder was 164.7 ft. The diameter of the cylinder varied from 2.299 in. on the *O* end to 2.324 in. on the *l* end. The deflection of the light beam initially tangent to the cylinder on the *O* end was determined by exposing sensitized paper on a screen *E* located 17 ft from the *O* end of the

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†Instructor of Mechanical Engineering, University of California.
[1]Schlierenaufnahmen des Temperaturfeldes in der Nähe wärmeabgebender Körper, by E. Schmidt (*Forschung*, Vol. 3, No. 4, 1932, p. 181-189).
[2]Heat Transmission, by W. H. McAdams (McGraw-Hill, 1933).
[3]Der Wärmeübergang an Rohren bei freier Strömung unter Berücksichtigung der Bildung von Schwitzwasser und Reif, by W. Piening (Gesundheits Ingenieur, No. 42, Oct. 1933).
[4]An Optical Method for Measuring Temperature Distribution and Convective Heat Transfer, by R. B. Kennard (*Bureau of Standards Journal of Research*, Vol. 8, June, 1932, No. 6, p. 787).
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Hot Springs, Va., June, 1938.
[5]Photographic Records of Heat Transfer, by S. Ruppricht (*Refrigerating Engineering*, Vol. 25-26, 1928).

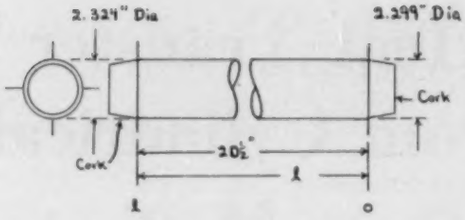

Fig. 2—Diagram of cylinder showing projecting needles

cylinder. The cylinder was chromium plated to reduce the surface emissivity and to preserve the surface.

The angle which the light beams made with the center of the cylinder was $2.30/2 \cdot 12 \cdot 164.7 = 0.000582$ radians. If the cylinder had not been tapered the light beams would have been $0.000582 \times 20\frac{1}{2} = 0.0121$ in. from the cold cylinder surface at the l end. The cylinder was so tapered that the light beam was tangent for the complete length of the cold cylinder. The diameter of shadow of the cold cylinder should have been

$$\frac{164.7 + 17.0}{164.7} \times 2.30 = 2.54 \text{ in.}$$

The measured shadow of the cold cylinder was slightly larger (0.06 in.). This effect was attributed to diffraction and to the penumbra due to the finite size of the source. The computed cold shadow size was used in the calculations.

The cylinder was aligned with the light beam by means of projecting needles on each end, as shown in Figs. 2, 3 and 4. The cylinder was considered to be aligned when the shadows of corresponding projecting needles coincided on screen E (Fig. 1). A support for the cylinder was so arranged that final alignment could be made by means of leveling screws.

The center of the cold shadow was located at the intersection of lines through the images of the needles.

As shown in the appendix, the slope of a light beam (initially tangent) as it left the cylinder was determined by the deflection from the shadow cast by the cold cylinder on the screen. The deflection was considered to be the distance from the computed shadow of the cold cylinder to the line of impingement of the outermost beam of light in the shadow from the hot cylinder. The deflection of the light beam was recorded by the exposure of sensitized paper at the screen E. (Examples of photographs obtained are shown in Figs. 5 and 6.)

For uniformity of temperature over the surface the cylinder was made from a piece of solid shafting with a ½-in. diameter hole drilled through the center in which an electrical heater was placed.

To show the effect of certain simply shaped surroundings, flat, water-cooled walls at various distances from the cylinder were arranged to form:

1. A vertical wall parallel to the cylinder.
2. A horizontal ceiling.
3. An intersection of a wall and ceiling parallel to the cylinder.
4. A tunnel, square in cross-section, located concentrically about the cylinder.

All flat surfaces were 3 ft long in a direction parallel to the cylinder. The room temperature varied from 75 to 78 F. The wall temperatures were 5 to 8 F below the room temperature.

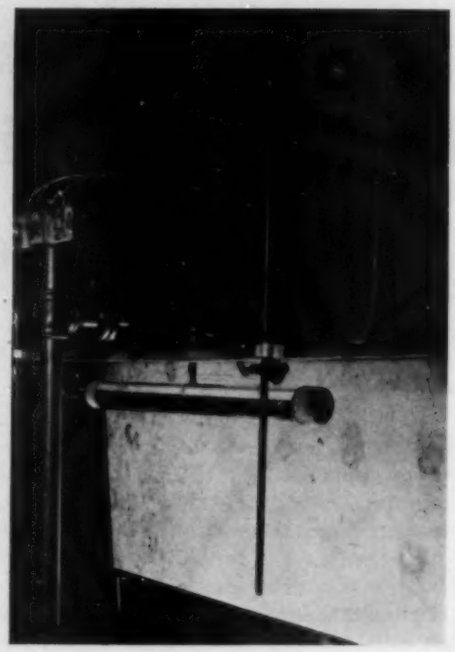

Fig. 3—View of cylinder arranged for adjustments

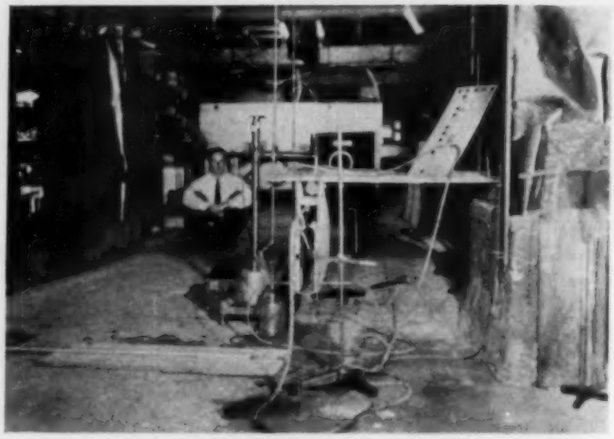

Fig. 4—General view of test apparatus

Evaluation of Experimental Results

The equation employed for the computation of the convective rate of heat loss, as indicated in Fig. 7 and appendix, was:

$$fd/k_{av} = N_H = \frac{k_w d}{k_{av}(T_w - T_f)} \left(\frac{T_w^3}{0.288}\right)^{\frac{1}{4}} \left(\frac{\beta^{\frac{1}{2}}}{\beta + 1} + tan^{-1}\beta^{\frac{1}{2}}\right) \quad (1)$$

where

$\beta = f^2 T_w / 0.288.$

$f = \frac{y_L - y_1}{l_1} \cong \frac{y_L}{L}$ = slope of light beam as it leaves the cylinder.

T_w = temperature of cylinder, degrees Fahrenheit absolute.

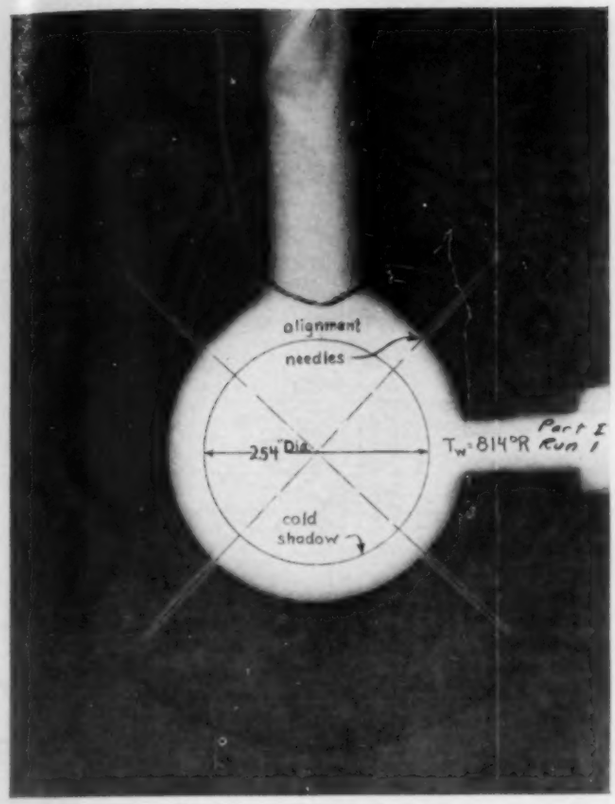

Fig. 5—Photographic record showing deflection of light beam for run 1

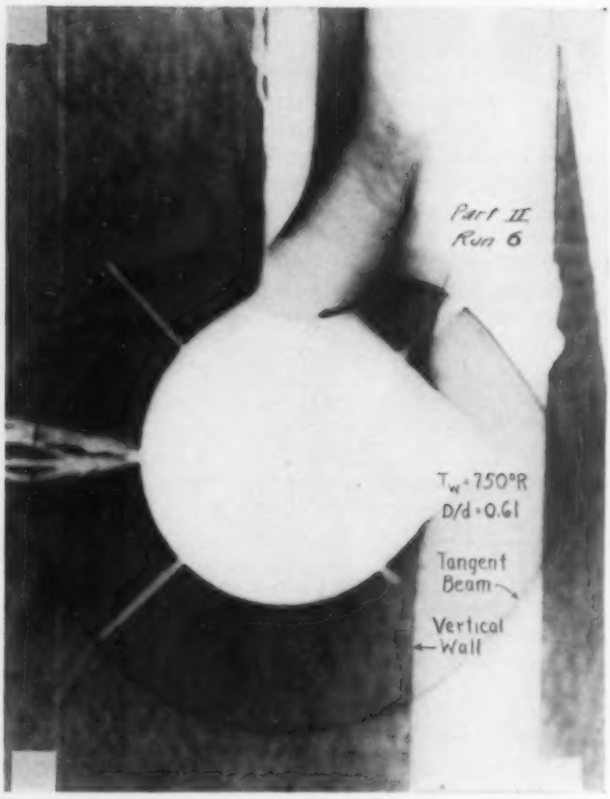

Fig. 6—Photographic record showing deflection of light beam for run 6

T_f = temperature of ambient air, room temperature, degrees Fahrenheit absolute.

y_L = deflection of tangent light beam at screen E, feet.

L = distance from center of cylinder to screen, feet.

y_1 = deflection of light beam as it leaves the cylinder, feet.

l = length of cylinder, feet.

$l_1 = L - l/2$.

k_w = thermal conductivity of air at temperature, T_w, Btu per (square foot) (hour) (degree Fahrenheit per foot).

k_{av} = thermal conductivity of air at the average temperature, $\frac{T_w + T_f}{2}$, Btu per (square foot) (hour) (degree Fahrenheit per foot).

d = cylinder diameter, feet.

f = convective film conductance per unit area, Btu per (square foot) (hour) (degree Fahrenheit).

The equation is based upon the following conditions:

1. For the path through which the tangent light beam travels, the air temperature is a linear function of the distance measured perpendicular to the cylinder wall.
2. The index of refraction, of air is given by the equation

$$n = 1 + \frac{0.144}{T}\,\frac{b}{760}$$

where

b = barometric pressure in millimeters of mercury.

T = air temperature, degrees Fahrenheit absolute.

3. The diffraction and penumbra effects are small.
4. The end conditions do not affect the light beam.

Equation (1) differs from that given by Schmidt[6] in that

[6]Loc. Cit. See Note 1.

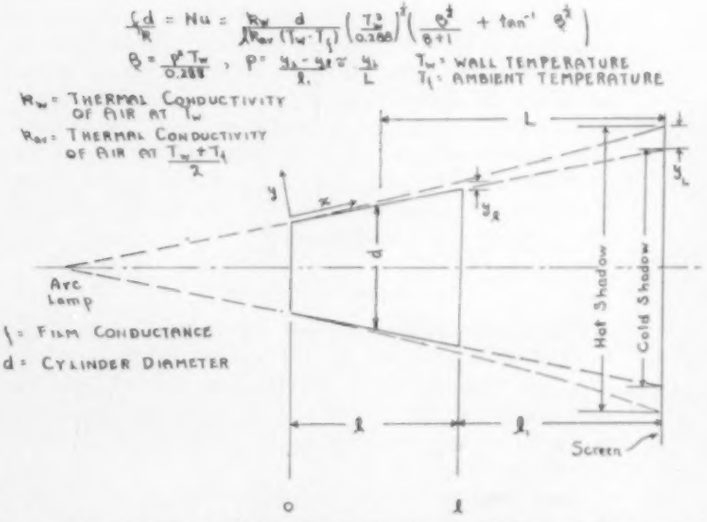
$\frac{fd}{k} = Nu = \frac{k_w}{k_{av}}\frac{d}{(T_w - T_f)}\left(\frac{T_w}{0.288}\right)^{\frac{1}{2}}\left(\frac{\beta^{\frac{1}{2}}}{\beta+1} + \tan^{-1}\beta^{\frac{1}{2}}\right)$
$\beta = \frac{p^2 T_w}{0.288}$, $P = \frac{y_1 - y_2}{l_1} = \frac{y_1}{L}$
T_w = WALL TEMPERATURE
T_f = AMBIENT TEMPERATURE
k_w = THERMAL CONDUCTIVITY OF AIR AT T_w
k_{av} = THERMAL CONDUCTIVITY OF AIR AT $\frac{T_w + T_f}{2}$
f = FILM CONDUCTANCE
d = CYLINDER DIAMETER
Arc Lamp
Hot Shadow
Cold Shadow
Screen

Fig. 7—Diagrammatic sketch of cylinder for developing computation of convective rate of heat loss

$\left(\frac{\beta^{\frac{1}{2}}}{\beta+1} + \tan^{-1}\beta^{\frac{1}{2}}\right)$ replaces $2\beta^{\frac{1}{2}}$.

The ratio $2\beta^{\frac{1}{2}} \Big/ \left(\frac{\beta^{\frac{1}{2}}}{\beta+1} + \tan^{-1}\beta^{\frac{1}{2}}\right)$ may be expressed approximately as $\eta = 1 + 0.62\,\beta$ (2)

In the experimental work summarized below η may be as large as 1.2.

Experimental Results

To justify the assumptions made in the derivation of equation (1), the values of film conductances determined for a horizontal cylinder placed in free space by the Schlieren method are compared with those obtained by other methods. In Fig. 8 the film conductances given in Table 1 are shown as a function of

$$\frac{\text{temperature difference}}{\text{cylinder diameter}}$$

and agreement is well within the accuracy of the data.

In Fig. 9 the data are compared with recommended values in terms of the generalized variables Nu and Gr/St,

where

Gr = Grashof's modulus = $\dfrac{gd^3\rho^2(T_w - T_f)}{\mu^2 T_{av}}$,

St = Stanton's modulus = 1/Prandtl's modulus = $\dfrac{k_{av}}{\mu g c_p}$

$T_{av} = \dfrac{T_w + T_f}{2}$

μ = viscosity of fluid, pound second per square foot.

ρ = mass density of fluid, pound (second)^2 per (foot)^4.

k_{av} = thermal conductivity of fluid, Btu per (square foot) (second) (degree Fahrenheit per foot).

c_p = specific heat of fluid at constant pressure, Btu per (pound) (degree Fahrenheit).

g = acceleration due to gravity, feet per second squared.

All fluid properties were determined at the mean film temperature and were taken from the International Critical Tables.

The experimental data agree well with the curve recommended by McAdams.[7]

The effects of four geometrically simple boundary conditions, given in Table 2, are indicated graphically in Fig. 10. The ratio of film conductance, with the cylinder at any distance from the walls, to that with the walls far from the cylinder, is shown as a function of the dimensionless ratio, the distance of cylinder center from wall/diameter of cylinder. The value of T_f in Grashof's Modulus and in equation (1) was taken as the room temperature. The correlations may not apply outside of the range of data investigated.

Acknowledgment

The authors wish to express their appreciation of the work of W. B. Chiswell, T. B. Klakring, W. Stahl, A. R. Joyce, F. Agens, and J. C. Broach, former graduate students, and D. N. Harris and T. R. Griffin, graduates of the University of California, Department of Mechanical Engineering.

These men aided in the development of the experimental technique and obtained the laboratory data.

[7]Loc. Cit. See Note 2.

Table 1—Film Conductance Values for Horizontal Cylinder

Run	T_w, Deg Fahr Abs.	T_f, Deg Fahr Abs.	Nu_{av}	f Btu per (Hour) (Sq Ft) (Deg Fahr)	$(Gr/St) \times 10^{-6}$	$\frac{(T_w - T_f)}{12d}$ Deg Fahr per Inch
1	814	537	16.7	1.43	1.17	119
2	700	535	16.0	1.29	1.04	71
3	759	535	16.3	1.35	1.13	96
4	800	535	16.7	1.43	1.17	114
5	856	535	16.8	1.47	1.18	138
6	935	535	17.5	1.60	1.15	172
7	652	535	15.0	1.15	0.72	50
8	831	535	16.7	1.46	1.17	127

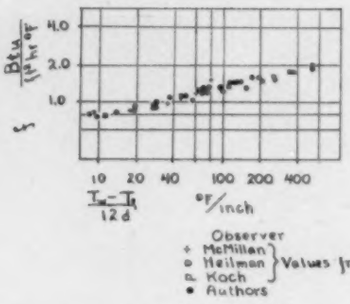

Fig. 8—Thermal free convection from a horizontal cylinder in free space

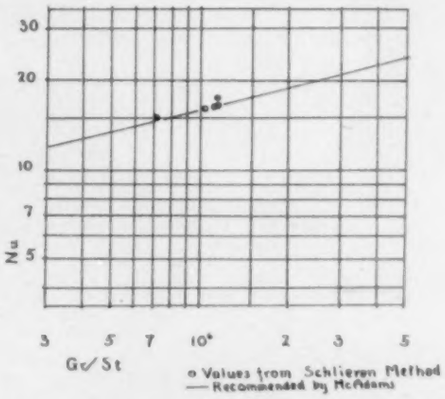

Fig. 9—Comparison of film conductance values in terms of Nu and Gr/St

Appendix

1. The index of refraction n of a medium may be defined by

$$Vn = C \tag{3}$$

where C is the velocity of light in a vacuum and V the velocity of light in the medium.

In Fig. 11 let R be the radius of curvature of a beam of light passing the surface $y = O$. The velocity of light at radius R is V and at $R + dR$, $V + dV$. In time $d\theta$ the beam at A travels a distance $V d\theta$, while ray at B travels a distance $(V + dV)d\theta$.

By similarity of triangles $\dfrac{1}{R} = \dfrac{1}{V}\dfrac{dV}{dR}$ (4)

Combining equations 3 and 4 gives

$$\frac{1}{R} = \frac{-1}{n}\frac{dn}{dt}\frac{dt}{dR} \tag{5}$$

If the ray is approximately parallel to the surface $y = O$,

$$\frac{1}{R} = \frac{d^2y}{dx^2} = \frac{+1}{n}\frac{dn}{dt}\frac{dt}{dy} = \frac{+m}{n}\frac{dn}{dt} \tag{6}$$

where, y = distance perpendicular to the surface,
x = distance parallel to the surface,

Table 2—Effects of Four Geometrically Simple Boundary Conditions

Vertical Wall		Ceiling		Corner		Tunnel	
D/d	f/f_∞	D/d	f/f_∞	D/d	f/f_∞	D/d	f/f_∞
0.93	0.992	1.20	1.0	1.26	1.0	1.56	1.0
0.83	1.008	0.93	1.0	1.04	0.988	1.15	0.970
0.77	1.000	0.83	0.953	0.93	0.975	0.93	0.905
0.72	1.001	0.72	0.915	0.83	0.949	0.83	0.867
0.66	1.003	0.66	0.858	0.72	0.933	0.72	0.857
0.61	1.005	0.61	0.837	0.66	0.864	0.67	0.752
0.55	0.963	0.56	0.815	0.61	0.826	0.61	0.706
				0.56	0.748		

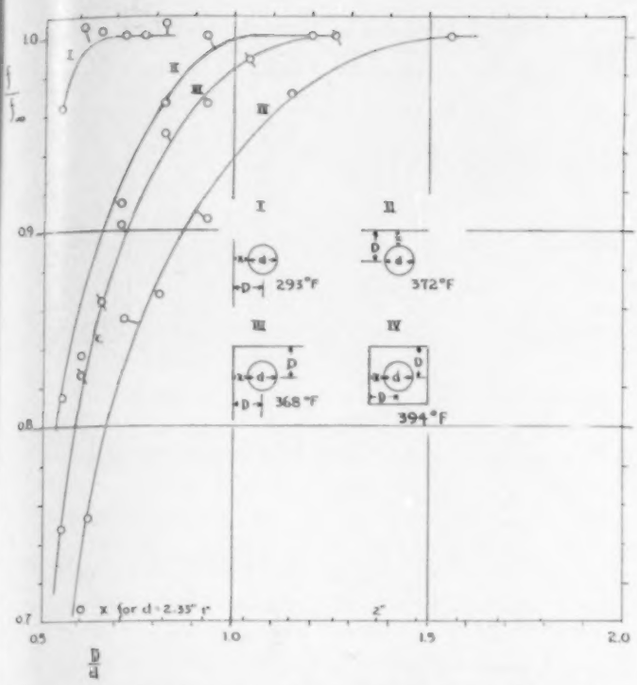

Fig. 10—Heat transfer by free convection from horizontal cylinders as a function of distance from surfaces variously arranged

$$m = \frac{dt}{dy},\ t = \text{temperature.}$$

For air, $$n = 1 + \frac{0.144}{T}\,\frac{b}{760} = 1 + \frac{A}{T} \qquad (7)$$

b = barometric pressure in millimeters mercury.
T = temperature, degrees Fahrenheit absolute.

$$A = 0.144\,\frac{b}{760}$$

Since n is within 0.03 per cent of being unity,

$$\frac{1}{n}\frac{dn}{dt} = \frac{-A}{T^2} \qquad (8)$$

Then $$\frac{d^2y}{dx^2} = \frac{-mA}{T^2} \qquad (9)$$

The boundary conditions for the integration are,

a. $x = O,\ \dfrac{dy}{dx} = O,\ y = O$

b. $x = l,\ \dfrac{dy}{dx} = p,\ y = y_1$

For convenience in computation, define the dimensionless quantity β as $\dfrac{p^2 T_w}{2A}$, T_w = surface temperature degrees Fahrenheit absolute.

2. If the cylinder is short, the surface temperature low, and the transfer factor low, the deflection, y_1, is small. Then T in equation (9) may be considered constant. With $T = T_w$, integration of equation (9) yields

$$\frac{dy}{dx} = \frac{-Amx}{T_w^2} \qquad (10)$$

$$p = \frac{-Aml}{T_w^2} \qquad (11)$$

$$\text{or } m = \frac{-pT_w^2}{lA}$$

$$m = \frac{-1}{l}\sqrt{\frac{\overline{T_w^2}}{2A}}\ 2\sqrt{\beta} \qquad (12)$$

$$= \frac{-1}{l}\sqrt{\frac{\overline{T_w^2}}{2A}}\ \phi_1(\beta) \qquad (12a)$$

With the thermal conductivity determined at the mean temperature of the tube and of the ambient air, and with the tube diameter as the significant length, Nusselt's modulus is

$$Nu = \frac{k_w d}{k_{av}(T_w - T_f)l}\sqrt{\frac{\overline{T_w}}{0.288}}\ \phi_1(\beta) \qquad (13)$$

3. For higher temperatures and longer cylinders the assumption $T = T_w$ in equation (9) is not valid. The temperature may be considered a linear function of distance out to 4 or 5 millimeters from the cylinder wall.[6]

Let $T = T_w + my$ in equation (9)

and $$\frac{d^2y}{dx^2} = \frac{-Am}{(T_w + my)^2} \qquad (14)$$

Integrate with the limits

$\dfrac{dy}{dx} = O$, for $x = O$, $y = O$, and $\dfrac{dy}{dx} = \dfrac{dy}{dx}$ at $x = x$,

which yields $$\frac{dy}{dx} = \sqrt{\frac{2A}{T_w}}\ \sqrt{\frac{1}{1 + my/T_w} - 1} \qquad (15)$$

Integration of (15) between the limits $y = O$ to y_1, $x = O$ to l yields $$m = \frac{-T_w}{l\sqrt{\dfrac{2A}{T_w}}}\left[(z^{\frac{1}{2}}(1-z)^{\frac{1}{2}} + \tan^{-1}\left(\frac{z}{1-z}\right)^{\frac{1}{2}}\right] \qquad (16)$$

where $z = \dfrac{-my_1}{T_w}$

From equation (15)

$$p = \sqrt{\frac{2A}{T_w}}\sqrt{\frac{1}{1-z} - 1} \qquad (17)$$

From equation (17)

$$z = 1 - \frac{1}{\beta + 1} \qquad (18)$$

With equation (18) in equation (16)

$$m = -\frac{1}{l}\sqrt{\frac{\overline{T_w^2}}{2A}}\left(\frac{\beta^{\frac{1}{2}}}{\beta + 1} + \tan^{-1}\beta^{\frac{1}{2}}\right) \qquad (19)$$

$$m = -\frac{1}{l}\sqrt{\frac{\overline{T_w^2}}{2A}}\ \phi_2(\beta) \qquad (19a)$$

Thus

$$f = \frac{-k_w m}{(T_w - T_f)} = \frac{k_w\sqrt{T_w}\,\phi_2(\beta)}{(1 - T_f/T_w)l\sqrt{2A}} \qquad (20)$$

and

$$Nu = \frac{k_w d}{k_{av}(T_w - T_f)l}\sqrt{\frac{\overline{T_w^3}}{0.288}}\ \phi_2(\beta) \qquad (21)$$

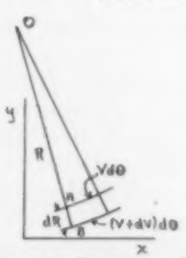

Fig. 11—Diagram of light beam travel

The values of $\phi_2(\beta)$ are given in Table 3 and Fig. 12. The function $\phi_2(\beta)$ approaches the function $\phi_1(\beta)$ as a limit as β approaches zero.

The ratio η of equation (12a) to equation (19a) indicates the error in employing equation (12a) for the determination

[6]The Transmission of Heat by Radiation and Convection by E. Griffiths and A. H. Davis (Dept. of Scientific and Industrial Research, Food Investigation Special Report, No. 9, 1931, rev. ed., H. M. Stationery Office, London).

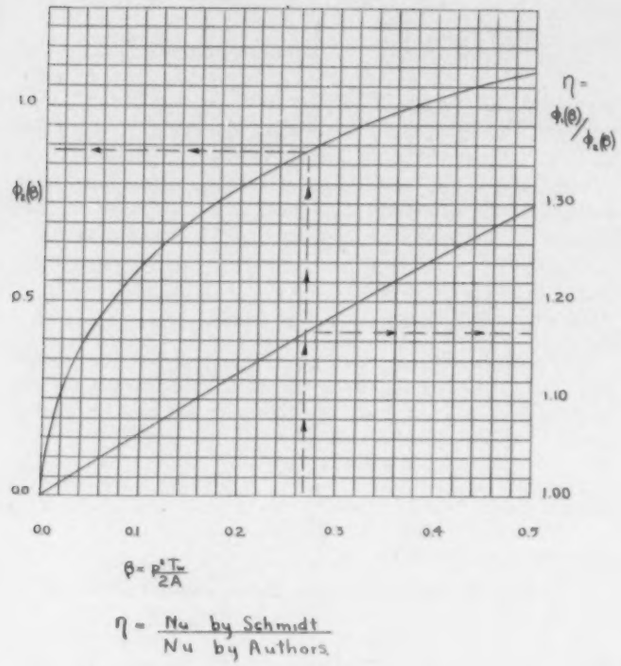

Fig. 12—Magnitudes of ϕ_2 (β) and η

of Nusselt's modulus. The values of η are shown in Table 3 and in Fig. 12. Approximately, η is a linear function of β and may be expressed by

$$\eta = 1 + 0.62\,\beta \tag{22}$$

As a close approximation,

Table 3—Tabulation of Convective Heat Loss Correction Factor (η)

β	$\phi_2(\beta)$	η	β	$\phi_2(\beta)$	η
0.0005	0.0448	1.000	0.04	0.390	1.026
0.001	0.0632	1.000	0.05	0.433	1.033
0.002	0.0892	1.002	0.10	0.594	1.065
0.003	0.1094	1.003	0.15	0.706	1.095
0.004	0.1262	1.003	0.20	0.793	1.126
0.005	0.1410	1.003	0.25	0.864	1.157
0.006	0.1543	1.004	0.30	0.923	1.188
0.007	0.1666	1.004	0.35	0.973	1.216
0.009	0.1885	1.006	0.40	1.016	1.243
0.01	0.1987	1.007	0.45	1.054	1.275
0.02	0.279	1.014	0.50	1.088	1.300
0.03	0.340	1.019	0.55	1.117	1.328
			0.60	1.143	1.355

$$Nu \cong \frac{k_w\, d}{k_{av.}(T_w - T_f)\, l} \sqrt{\frac{\overline{T_w}^2}{0.288}}\, \phi_1(\beta) \Big/ (1 + 0.62\,\beta)$$

In the computation by either method, the slope p must be known. Exactly,

$$p = \frac{y_L - y_1}{l_1} \tag{23}$$

As an approximation, with $T = T_w$, equations (10) and (11) yield

$$y_1 \cong \frac{p}{2} l \tag{24}$$

Since y_1 is small compared to y_L, a small error in y_1 will cause a small error in the computation of p by equation (23).

Substituting equation (24) into equation (23) yields (Fig. 4)

$$p = \frac{y_L}{l_1 + l/2} = \frac{y_L}{L} \tag{25}$$

Equation (25) was employed for the determination of slope, p. The error due to this approximation may be two or three per cent. The actual value of p may be found by trial and error methods if greater accuracy is desired.

Research Executive Committee Meets at New Haven

The Research Executive Committee held a meeting in the laboratory of the John B. Pierce Foundation at New Haven, Conn., on September 17. Chairman W. L. Fleisher, New York, presided and the following members were in attendance: J. H. Walker, Detroit, Prof. C.-E. A. Winslow, New Haven, C. Tasker, Toronto, Lt. Col. W. A. Danielson, Louisville, Ky., Director F. C. Houghten, Pittsburgh, John James, New York, and by invitation J. C. Fitts, New York, chairman of the Research Technical Advisory Committee on Radiation and Comfort Winter and Summer and Effect of Varying Humidity on Radiant Heating and Cooling.

Before calling the meeting to order Professor Winslow personally escorted the members through the various rooms in the laboratory and described research investigations now being conducted.

A progress report was made by Director Houghten regarding the results of the comfort studies conducted this summer at the Metropolitan Life Insurance Building in New York City, the Federal Reserve Building, Washington, D. C., and the Public Service Building in San Antonio, Tex. Preliminary papers covering this work are expected to be completed in time for presentation at the Annual Meeting of the Society in Pittsburgh next January.

Other important matters discussed at the meeting included consideration of the construction of two additional test rooms to be built at the Research Laboratory in Pittsburgh; approval of the initiation of a cooperative agreement at the University of Illinois to conduct additional air distribution tests; and a review of the possibilities of expanding the observations which were made in Pittsburgh this summer for the Committee on Air Conditioning in Industry.

Air Cleaning Committee Holds Meeting

A general meeting of the Research Technical Advisory Committee on Air Cleaning and Atmospheric Impurities was held in Washington, D. C. on September 30 to discuss the revision of the Society's Standard Test Code for Air Cleaning Devices and other subjects relating to the control of air borne organisms on the effect of air pollution and disease.

The chairman of the Committee is H. C. Murphy, Louisville, Ky., and the personnel of the Committee includes Dr. Leonard Greenburg, J. J. Bloomfield, W. H. Carrier, R. S. Dill, Theodore Hatch, C. E. Lewis, A. L. Simison, W. O. Vedder, Dr. E. B. Phelps, H. E. Adams, R. D. Bennett, L. W. Chubb, L. R. Koller, G. W. Penney, W. F. Wells.

Seasonal Variations in Effective Temperature Requirements

By F. E. Giesecke* and W. H. Badgett** (*MEMBERS*), College Station, Texas

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Texas Agricultural and Mechanical College, conducted under the supervision of the Research Technical Advisory Committee on Sensations of Comfort.

COMFORT requirements for air conditioning have been the subject of extended research by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory and at cooperating institutions since 1923. Early in this work it became apparent that standards of comfort, insofar as temperature, humidity, and air movement are concerned, are not absolute, but are considerably affected by climatic and other conditions. This resulted in the subsequent development of a summer and a winter comfort zone.

These comfort zones at first were assumed to apply universally to indoor air conditioning, regardless of daily variations in the outdoor temperature. In applying these findings to air conditioning in theaters and elsewhere, it soon became apparent, however, that the information available was inadequate and that some consideration should be given to outside temperature. Consequently, there was developed a variable indoor summer comfort standard based upon the outdoor dry-bulb temperature, which was used in the editions of 1933 to 1937 of THE A.S.H.V.E. GUIDE. This standard was based upon a single definite moisture content or dew-point temperature of the air, regardless of the outdoor conditions, and a definite indoor dry-bulb temperature for any given outdoor dry-bulb temperature. Attempts in employing this standard in practice for different air conditioning applications and with different types of equipment resulted in considerable difficulty and dissatisfaction, and the consequent resubmission of the study to the Society's Research Laboratory. In view of the Laboratory's findings in 1935 and 1936, these standards were revised in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1938 and are based upon effective temperature in the conditioned space in relation to the outdoor temperature rather than upon a constant moisture content of the indoor air.

In the Society's summer comfort study in Pittsburgh[1] in 1935 and 1936 it was indicated that an effective temperature of about 72.5 deg was desirable for maximum summer comfort rather than the 66 deg ET generally accepted for winter heating and air conditioning.

Yaglou and Drinker[2] found at Boston that there was an apparent relation between the optimum temperature for comfort which seemed to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained approximately the same value for the months of July, August, and September, 1927, and a decrease in the optimum temperature became apparent only when the prevailing outdoor temperature dropped to 66 F, which is below the customary room temperature in the United States for summer and winter. They found that the average optimum temperature for comfort was about 4.5 deg ET higher in summer than that required in winter, a variation which was ascribed partly to difference in clothing worn in the two seasons.

From these various studies it became apparent that there must be some gradual transition from the winter comfort zone to the summer comfort zone, and vice versa, related in some manner to the outdoor temperature, daily, weekly, monthly, or seasonal average. In order to determine this seasonal variation in effective temperature requirements, if possible, a controlled laboratory study was undertaken by the Texas Engineering Experiment Station at the Agricultural and Mechanical College of Texas in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS which ran continuously from January 1937 through May 1938.

These studies were made in an air conditioned laboratory of the Texas Engineering Experiment Station at College Station, Texas. The room was 9 ft wide, 8 ft high, and 18 ft long, with all walls, floor, and ceiling insulated with 3 in. of cork-board. The walls were plastered, and, including the ceiling, were finished with aluminum paint. The floor was 1 in. pine laid over the cork.

The ceiling, which formed the lower side of a plenum chamber, was of rigid insulation made from exploded wood-fiber perforated with $\frac{5}{16}$ in. holes about $2\frac{1}{2}$ in. on centers. Into this plenum chamber the conditioned air was introduced and maintained at a slight static pressure which forced it through the ceiling perforations uniformly over the entire test room. Air was returned to the conditioner through two slotted grilles in the ceiling extending across each end of the long side of the room. These slots were arranged so that air was picked up uniformly across the width of the room.

A spray-type air washer, equipped with both cooling and reheating coils, was used to condition the air introduced into the room. A slow speed fan moving about 325 cfm was employed to circulate the conditioned air,

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*Director, Texas Engineering Experiment Station.

**Research Assistant, Texas Engineering Experiment Station.

[1]Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker, and Carl Gutberlet (A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936, p. 681).

[2]The Summer Comfort Zone: Climate and Clothing, by C. P. Yaglou and Philip Drinker (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 269).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

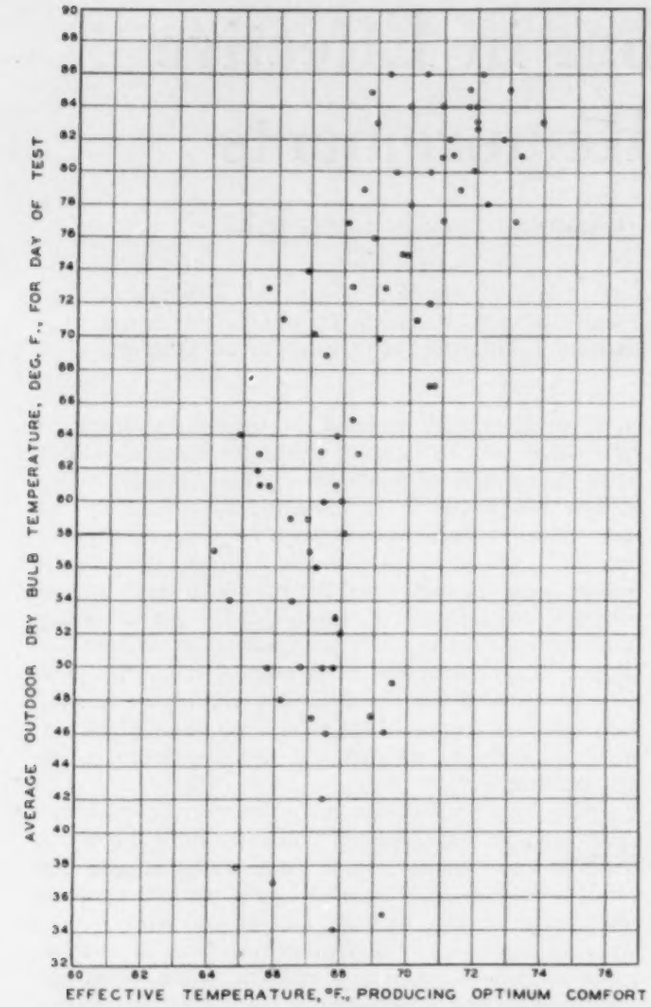


Fig. 1—Effective temperature producing optimum comfort plotted against average outdoor dry-bulb temperature for day of test

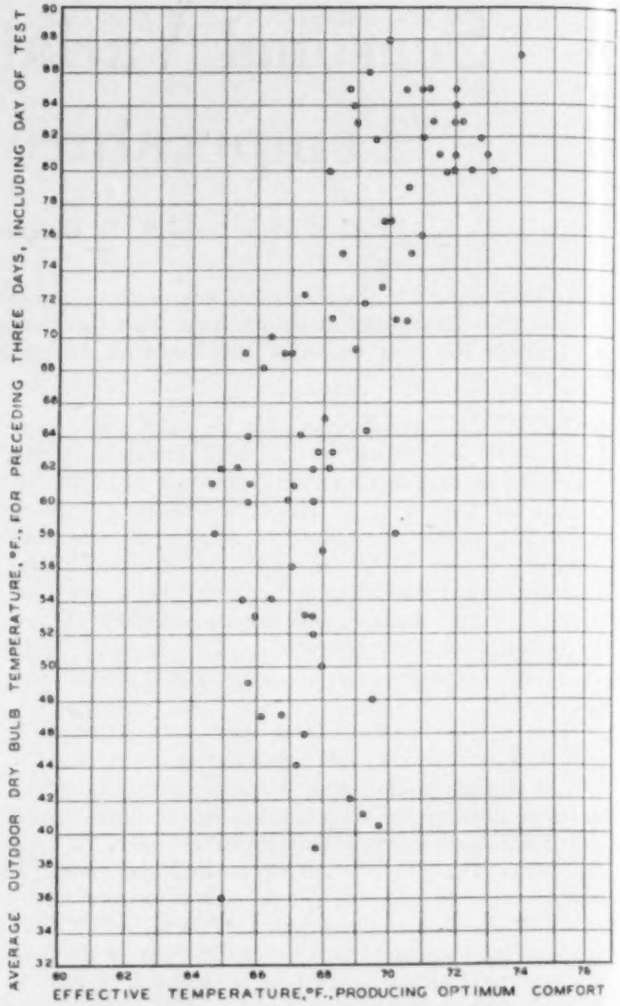

AVERAGE OUTDOOR DRY BULB TEMPERATURE, °F., FOR PRECEDING THREE DAYS, INCLUDING DAY OF TEST
EFFECTIVE TEMPERATURE, °F., PRODUCING OPTIMUM COMFORT

Fig. 2—Effective temperature producing optimum comfort plotted against average outdoor dry-bulb temperature for day of test and preceding two days

and, by means of this slow speed fan and the use of the perforated ceiling as a distribution grille, a very uniform and imperceptible air movement was obtained throughout the conditioned space, which was well within the meaning of the term *still air*. Fresh air was introduced at a rate of about 8 to 10 cfm per person.

The test procedure was similar to that employed in the previous comfort zone work of the Society. With 10 male students between the ages of 18 and 23 years, all in good health, serving as subjects, two tests were made each week, one on Thursday afternoon with a group of five subjects, and the other on Friday afternoon with the other five subjects. By using two groups of five subjects each, it was possible to have about 30 sq ft of floor space per subject, thus keeping them far enough apart so that inter-radiation of body heat was reduced to a minimum. The subjects were seated so that each was at least 12 in. from the nearest wall and at least 30 in. from his nearest neighbor. The lighting arrangement of the room was such that the effect of radiant heat from that source was eliminated.

The usual practice was to bring the subjects to equilibrium in a pre-determined constant condition in the test room, which, in each case, was *too warm* for comfort, after which the conditions were made to vary slowly across the comfort zone during a period of from three and one-half to five hours, at a rate never exceeding 1 deg in 8 to 10 min to a condition of *too cool* for comfort, while the subjects recorded their reactions to the conditions at intervals of 5 min according to the following scale of comfort which has been used in most of the Society's previous work on the comfort zone:

1. Cold.
2. Too cool for comfort.
3. Comfortably cool, *i.e.*, the subject was not particularly uncomfortable, but would for the sake of comfort desire a slightly warmer rather than a slightly cooler condition.
4. Ideally comfortable in regard to the sensation of warmth.
5. Comfortably warm, *i.e.*, the subject was not particularly uncomfortable, but would for the sake of comfort desire a slightly cooler rather than a slightly warmer condition.
6. Too warm for comfort.
7. Hot.

At the same time the degree of sensible perspiration was observed according to the following scale:

0. Forehead or body, dry.
1. Forehead or body, clammy.
2. Forehead or body, damp (perspiration just visible).

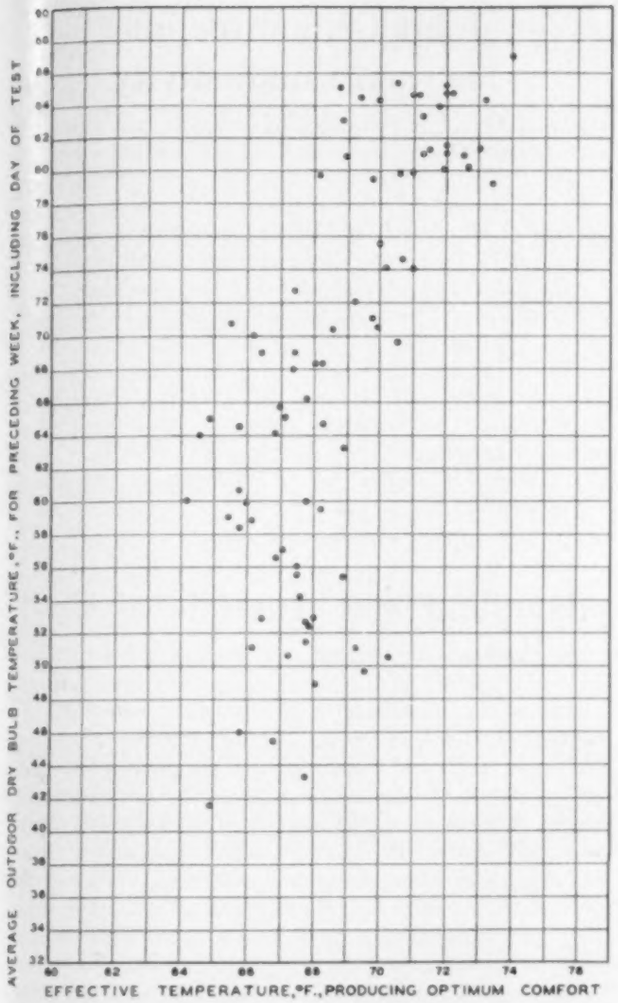

AVERAGE OUTDOOR DRY BULB TEMPERATURE, °F., FOR PRECEDING WEEK, INCLUDING DAY OF TEST
EFFECTIVE TEMPERATURE, °F., PRODUCING OPTIMUM COMFORT

Fig. 3—Effective temperature producing optimum comfort plotted against average outdoor dry-bulb temperature for the preceding week, including day of test

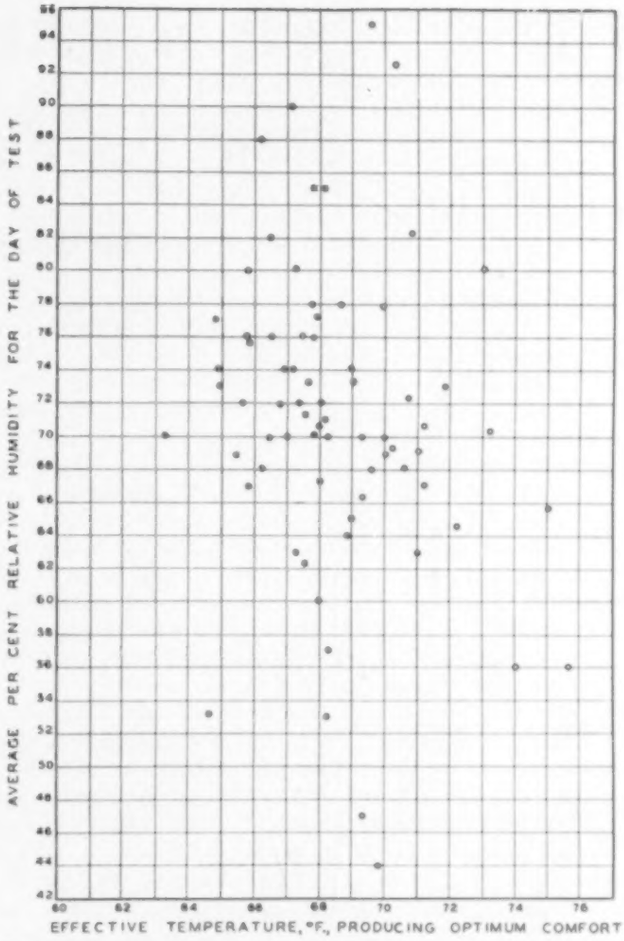


Fig. 4—Effective temperature required for optimum comfort plotted against average outdoor relative humidity for day of test

3. Forehead or body, wet (sweat covering the surface, frequently in drops).

4. Perspiration on the head runs down, or perspiration on the body runs down or wets the clothing.

All subjects were dressed similarly and wore the same clothing throughout all the tests, consisting of the following items: a medium weight wool suit without waistcoat, cotton shirt with collar and tie, cotton undershirt and cotton drawers, both of the type commonly worn in the south the year round, light weight standard length cotton socks, and low-top shoes. The total weight of the clothing worn by each subject, not including shoes, averaged about 4.4 lb, of which the suit averaged about 3.3 lb.

Three separate studies were made of the data secured from these tests. In each case the effective temperature producing optimum comfort for each group for each test was plotted against, first, the average outdoor dry-bulb temperature for the day of the test; second, the average outdoor dry-bulb temperature for the day of the test and the preceding two days; and, third, the average outdoor dry-bulb temperature for the preceding week including the day of the test. The effective temperature producing optimum comfort was considered as being the mean of the upper and the lower limits of zone 4 or the *Ideally Comfortable* zone. These data are shown in the Figs. 1, 2, and 3 respectively. Each point represents the average of the reactions of a group of five subjects for one afternoon's test. If the three figures are superimposed upon each other, they will be found practically identical in their general trend, although the individual points themselves do not coincide. It will be noted that above 65 F a marked relation is indicated between the outdoor dry-bulb temperature and the effective temperature required for optimum comfort and below 65 F these optimum conditions seem to remain constant or independent of the outdoor dry-bulb temperatures. This change in relationship occurs at the outdoor temperature below which it is generally conceded that heating is required for comfort.

This variation in relationship may be explained as follows: In the winter places of occupancy are closed to the outdoor weather; the greatest part of the time people are indoors and exposed to a more or less uniform temperature; and when they are outdoors, they are dressed according to the prevailing temperature, and, therefore, their response to variations in effective temperature requirements in relation to the outdoor temperature is reduced to a minimum, and, consequently, winter comfort requirements remain more or less constant throughout the season.

On the other hand, in summer, as it gets warmer, houses are opened and their temperature is allowed to fluctuate with the outdoor temperature; people dress as lightly as possible, and, consequently, whether they are indoors or outdoors, their bodies are exposed to temperatures following closely those of the outside, and, therefore, effective temperature requirements vary more closely with the outdoor temperatures.

In the area of the charts where the average outdoor temperature is above 78 or 80 F, above which temperature summer air conditioning or cooling is generally desired, it will be seen that the upper and lower limits of the comfort zone will extend from about 69 to 74 deg ET, which agrees with the summer comfort zone as previously established for the conditions obtaining at College Station. The optimum condition of maximum comfort for outdoor temperatures above 80 F would vary from about 70.5 deg ET to about 73 deg ET, a mean of about 71.7 deg ET, which is only slightly higher than the previously determined summer comfort line of 71 deg ET for the eastern part of the United States, as published in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, and is the same as the mean of the upper and lower limits of the comfort zone as established for Pittsburgh in the summer of 1936.

The line of optimum comfort of 67 deg ET for the winter comfort zone is only 1 deg ET above the 66 deg ET adopted by the Society as the winter comfort line. These variations between the effective temperature requirements for the North and East and South are so small that they are of no practical importance, particularly in view of the well established fact that comfort requirements vary through a zone rather than along a line.

To determine what relation may exist between the outdoor relative humidity and the optimum comfort requirements, Fig. 4 was drawn. In this chart the effective temperature required for optimum comfort is plotted against the average outdoor relative humidity for the day of test. No correlation between the two is apparent.

A study of the condition at which sensible perspiration disappeared was made and, for the 82 tests, it was found that the average effective temperature at which perspiration ceased was between 75 and 76 deg ET, which agrees with the previous work of the Society establishing this value as about 76 deg ET.

Additional tests were made to determine the variation in effective temperature requirements caused by the coats worn by the subjects during the tests. The finding of the Laboratory, that the wearing of a coat increased the effective temperature requirements approximately 2 deg, was verified.

Although these results are based upon determinations secured from only 10 subjects, all men and in good health, it is believed that they are conclusive enough to indicate that there is a definite relation between the indoor effective temperature required for comfort and the outdoor temperature which will be applicable to the general public; that it is indicated that comfort requirements remain more or less constant when the outdoor temperature falls below 65 F, and that above 65 F, the effective temperature required for optimum comfort varies with the outdoor dry-bulb temperature.

Joint Committee on Thermal Conductivity

A Joint A.S.H.V.E.-*A.S.T.M.* Committee has just been appointed to develop a Code for Determining the Conductivity of Materials. The Chairman of the Committee is Director F. C. Houghten of the Research Laboratory in Pittsburgh and other Society representatives are Prof. F. B. Rowley, Minneapolis, Minn., and Prof. G. B. Wilkes, Cambridge, Mass. The *American Society for Testing Materials* is represented by H. C. Dickinson, Washington, D. C., Prof. E. R. Queer, State College, Pa., and R. H. Heilman, Pittsburgh, Pa. Invitations were extended to the *American Society of Refrigerating Engineers* to cooperate with the project and they appointed C. B. Bradley, Manville, N. J., and the *National Research Council* selected H. Harrison, Chicago, Ill.

Sampling and Analysis of Coal

A report containing much material of value to chemists and engineers concerned with sampling, analysis, classification, or utilization of coals has been published by the Bureau of Mines, U. S. Department of the Interior, as Technical Paper 586, Notes on the Sampling and Analysis of Coal. The report is a revised and enlarged edition of a technical paper bearing the same title published in 1914.

Many of the methods of analyzing coal, especially the proximate analysis, are empirical in nature. The accuracy and limitations of these methods are explained and their relationships to the commercial value of coal are discussed. The Bureau of Mines has developed most of the methods for analyzing coal and coke now accepted as standard in this country and the publication summarizes many experimental data upon which these methods are based.

In addition to covering the analysis of coal and the laboratory preparation of samples for analysis, the publication discusses coal ash fusibility and its relation to clinker formation and the classification of coals by rank, that is, according to their degree of metamorphism. The paper is not intended as a laboratory manual for analysis of coal, but rather for showing the significance of analytical results and their bearing on use properties of coals.

Elected to Life Membership

K. Sekido
Tokyo, Japan
Elected 1903

J. K. Peacock
New York, N. Y.
Elected 1921

Edgar Herring
London, England
Elected 1919

The Circulation of Air in Scale Model Rooms

By E. C. Lundquist*, Buffalo, N. Y.

Introduction

THE investigation of the circulation of air in a room by the use of a scale model presents a problem that is of both theoretical and practical interest. It is well known how important and how practical the use of scale models has become in predicting the performance of airplanes, boats, hydraulic structures, and many other devices dependent upon fluid flow in their action. If it was possible to predict the circulation of air in any full-sized room by experimenting with a model built geometrically to scale, a great deal of expense and uncertainty could be eliminated in arranging the air distribution system to obtain satisfactory circulation. The purpose of this investigation was not only to find out the manner in which air circulated in a scale model room for various duct arrangements, but also to show that the air distribution was not substantially altered, for any particular duct arrangement, by changing the scale of the model, if dynamic similarity was maintained.

The requirement of similarity is fulfilled in a scale model experiment of this nature, assuming geometric similarity, if Reynolds' number in the inlet and outlet ducts to the scale model room is the same as in the prototype. Reynolds' number is defined by

$$Re = Dw\rho/\mu$$

where D is the characteristic linear dimension in feet,
w is the velocity of flow in feet per second,
ρ is the mass density of the fluid in slugs per cubic foot,
μ is the absolute viscosity in pounds seconds per square foot.

Experimental Procedure

A full-sized room was not available for experimental purposes, so two geometrically similar models of a room 12 ft by 12 ft by 10 ft high were built to scales of 1/10 and 1/5. To demonstrate the complete reliability of a scale model experiment in the circulation of air in a room it would have been valuable to compare the results obtained from a scale model to those obtained from the full-sized prototype. In this case, however, conclusions had to be drawn from the two scale models used.

For simplicity, circular ducts were assumed for the prototype. Furthermore, assuming a typical velocity in the duct leading to the inlet register of 700 fpm and that 16 air changes per hour were required, the quantity of air required and then the duct diameter for the prototype were calculated. The duct diameter turned out to be 10 in.; and since the models were built geometrically to scale, the duct diameters for the 1/10 and the 1/5 scale models were made respectively one inch and two inches. Making appropriate assumptions as regards the density and the temperature of the air in the inlet duct of the prototype, Reynolds' number in that duct was calculated. Assuming no appreciable change in density and viscosity between the conditions under which the prototype and the scale models operated, the Reynolds' numbers of the scale models were made to equal that of the prototype merely by increasing velocity of flow within the ducts by the reciprocal of the scale of the model. The velocity of flow within the ducts leading to the 1/5 scale model room was maintained at 3500 fpm for all experimental runs and at 7000 fpm for the 1/10 scale model room.

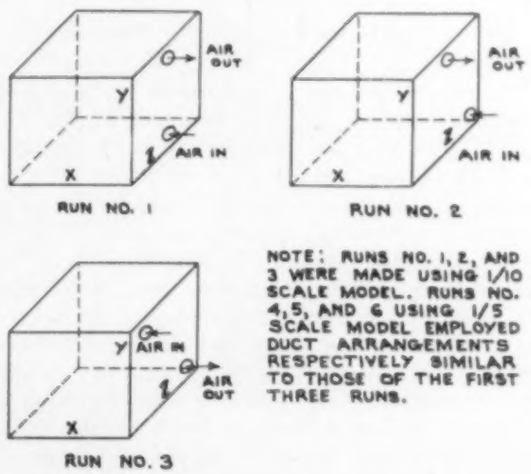


Fig. 1—Locations of inlet and outlet ducts for runs No. 1 to 6

Both the models were so constructed that the locations of the inlet and outlet ducts could readily be changed. In making comparative explorations of the air movements within each of the two models, the inlet and outlet ducts were similarly located. In this article the results for three different duct locations for each of the two models will be discussed. Fig. 1 shows the location of the ducts for Runs No. 1, 2, and 3 using the 1/10 scale model room. The location of the ducts for Runs No. 4, 5, and 6 were respectively similar to those for Runs No. 1, 2, and 3 but for these runs the 1/5 scale model room was used. For convenience in reference, the

*Buffalo Forge Company.

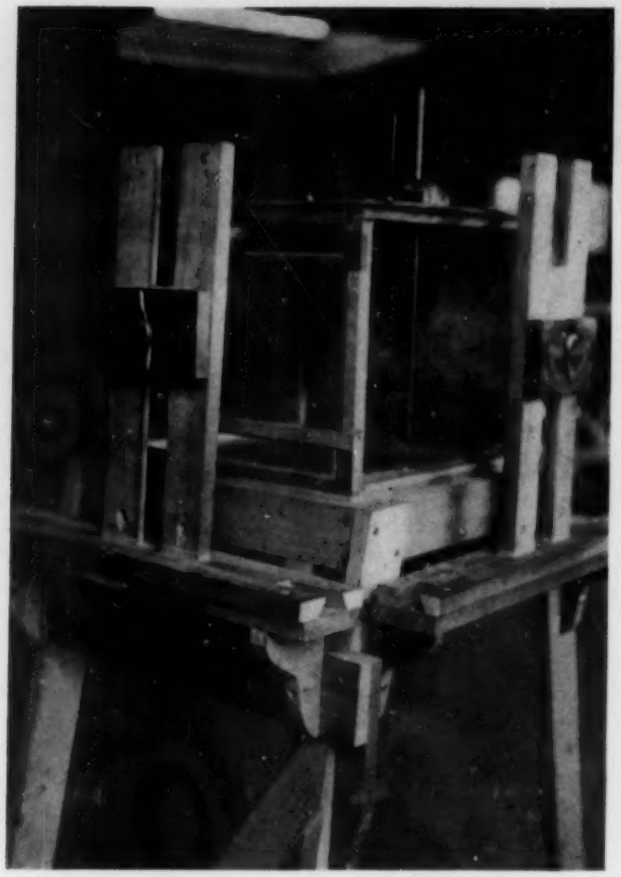


Fig. 2—Photograph of 1/10 scale model room

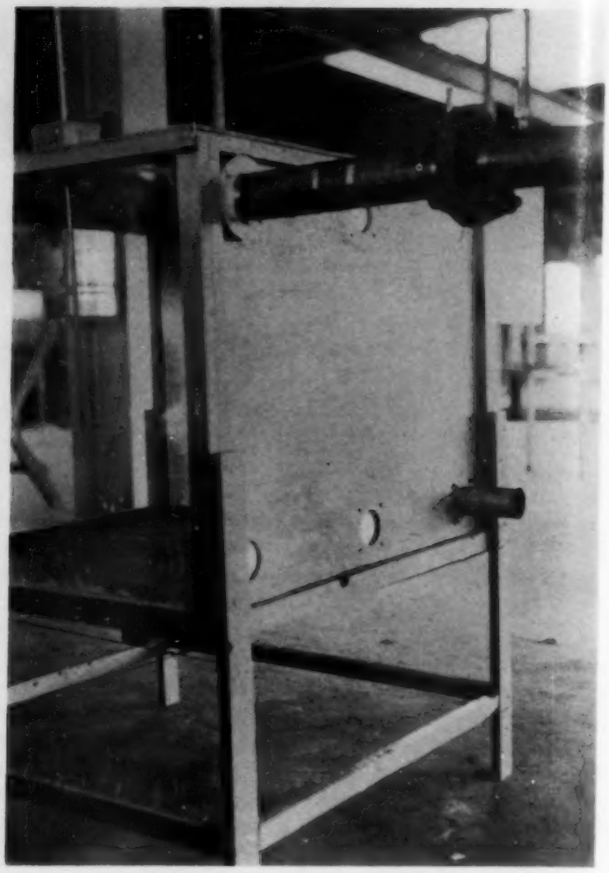


Fig. 3—Photograph of 1/5 scale model room

three mutually perpendicular corners of the room are labeled as the x, y, and z axes as shown in Fig. 1.

Each of the two scale models was built with at least two walls of glass to permit observations being made within. The velocities of air movement within the models were not measured, the direction of air flow being the object of investigation. The direction of air flow at various points throughout the room was obtained by observing the deflection of a fine silk tuft hung on a supporting rod extending down from the ceiling. Fig. 2 is a photograph of the 1/10 scale model room and shows the rod supporting the silk tuft. This rod was fitted into a pyramid base screwed on the roof of the ceiling in such a manner that it could be moved up or down. The ceiling itself was divided off into 16 stations such that the pyramid base holding the exploring rod could be fitted onto the ceiling at any one station. By sliding the rod up and down and by moving the rod from station to station, the end of the rod supporting the silk tuft could be made to occupy different positions throughout the room.

Since the silk tuft was exceedingly light, it could be depended upon to follow the direction of air movement at the point inside the room at which it was suspended. The direction of air movement at any one point was completely determined by observing the angularity of the tuft from two directions at right angles to one another. This was accomplished by the use of a pair of sliding protractor devices as shown photographically in Fig. 2.

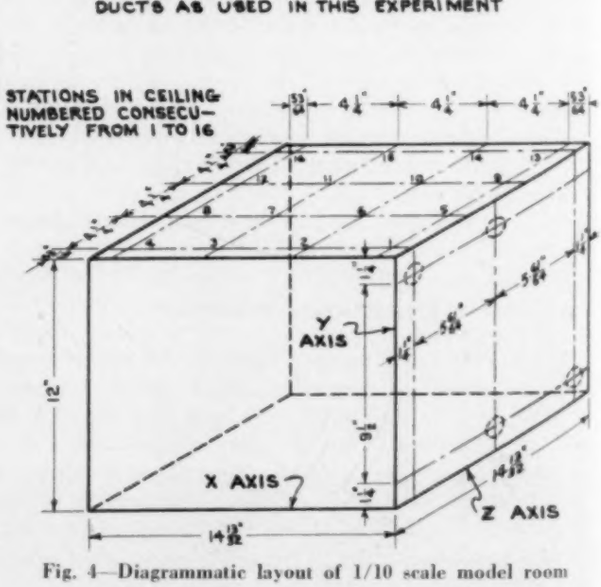


Fig. 4—Diagrammatic layout of 1/10 scale model room

Air was supplied to the inlet duct by a fan capable of delivering the desired quantity of air at the required static pressure. The air velocity within the duct leading to the 1/10 scale model was measured by means of a piezometer ring in combination with an impact tube, the device being calibrated before being used. The air veloc-

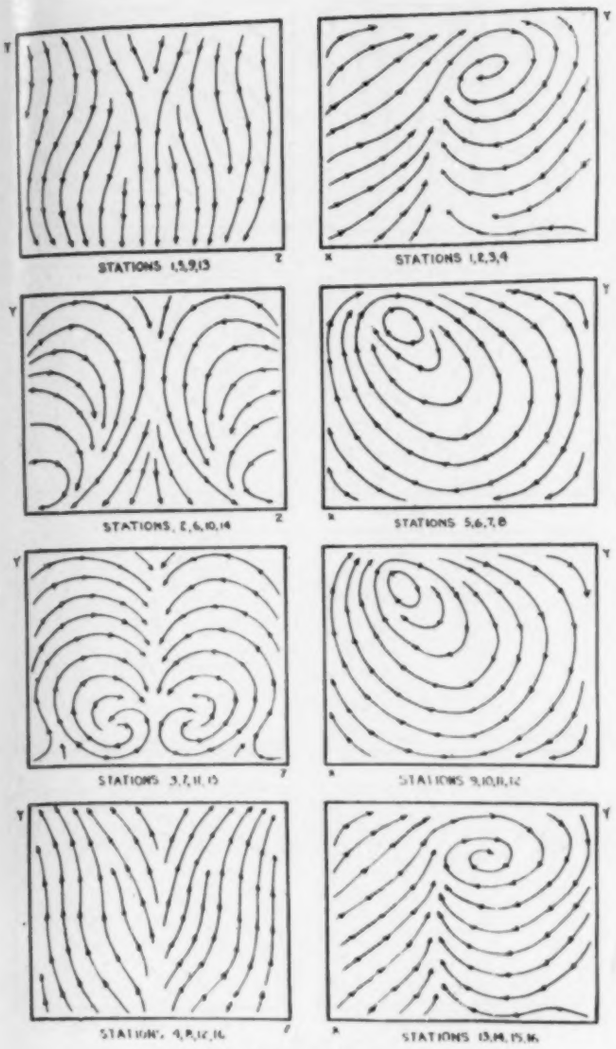


Fig. 5—Run No. 1

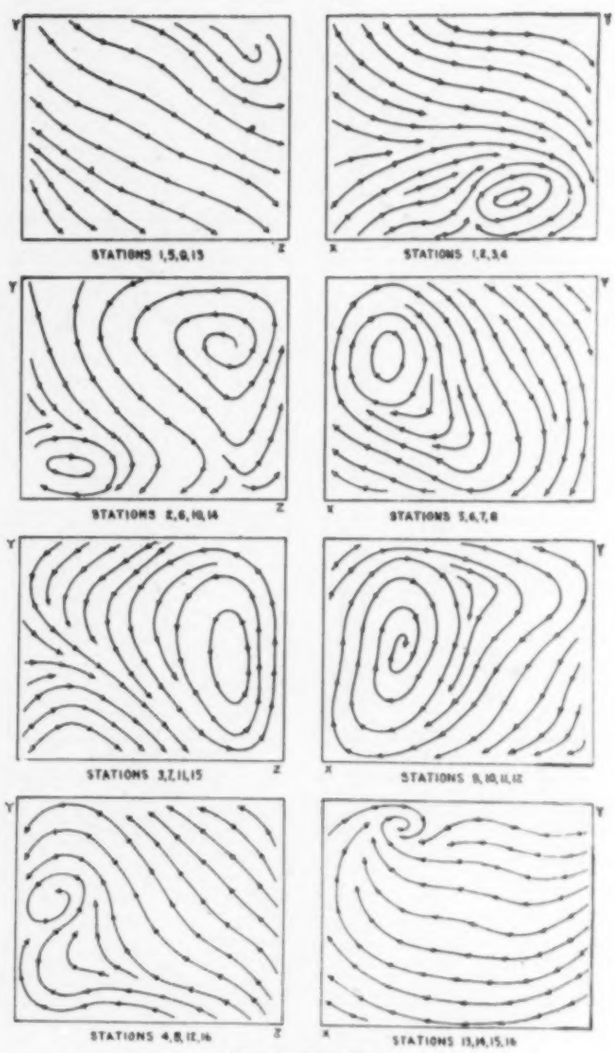


Fig. 6—Run No. 2

ity in the duct leading to the 1/5 scale model room was measured by an orifice inserted in that duct. The orifice was made to specifications given in the *A.S.M.E.* Fluid Meter Report of 1931 and the coefficient as given by this report was used in calculating the velocity.

Plotting the Air Streams

Sixteen different measuring stations were provided in a plane parallel to the ceiling and the floor in each of the models used in this experiment. At each one of these stations the distance between the floor and the ceiling was traversed by taking six observations of the deflection of the silk thread hanging from the end of the rod extending down from the ceiling. The direction of the air flow was then observed at 96 different points throughout the room. The traversing points at any of the stations were similarly located along a line joining the floor and the ceiling in each of the two models. That is, observations were made at points throughout the 1/5 scale model room which were similarly located to the points of observation in the 1/10 scale model room.

It would be difficult to clearly depict the actual conditions of flow within the model by means of a three dimensional plot. The simplest way of presenting the data was by plotting the flow across a series of mutually perpendicular vertical planes, and it was convenient to choose these planes as those passing through the measuring stations. Fig. 4 indicates the location of the measuring stations in relation to the coordinate axes for the 1/10 scale model room. All the dimensions would be doubled for the 1/5 scale model room.

In the air distribution diagrams that follow, each plot represents the flow across that vertical plane passing through the station points indicated. The labeling of the axes on the diagrams (Figs. 5 to 10 inclusive) indicates to which of the coordinate planes the section is parallel. It should be understood that the flow is not necessarily parallel to the plane of the diagram, the patterns merely representing the component of flow in that particular plane.

Fig. 1 shows the locations of the inlet and outlet ducts for the various test runs while the corresponding air distribution diagrams are shown in Figs. 5 to 10 inclusive.

Qualitative Description of Results

Test Run No. 1. Both the inlet and the outlet ducts for this test run using the 1/10 scale model room were

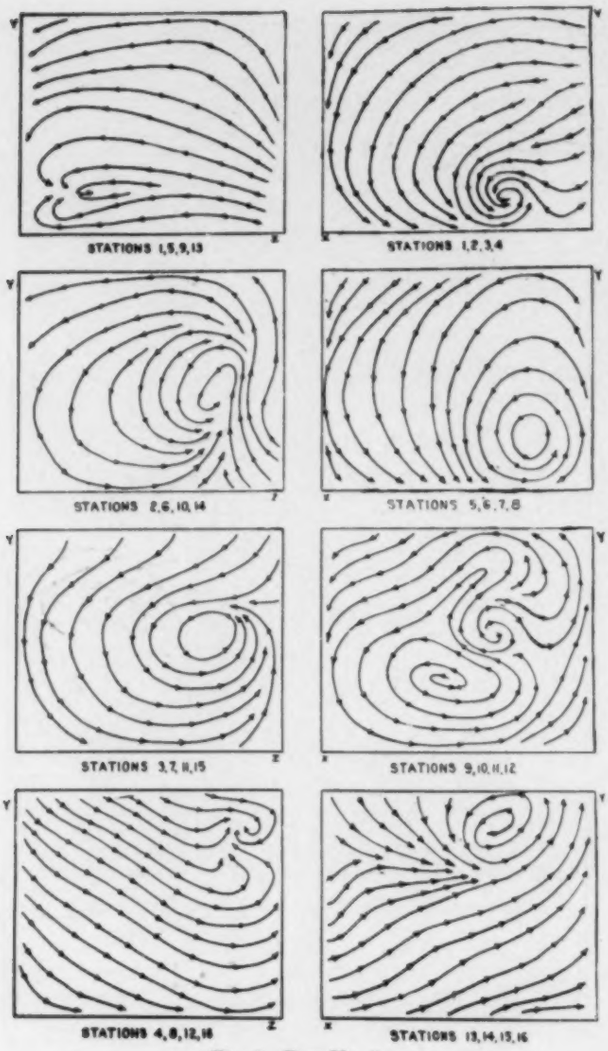


Fig. 7—Run No. 3

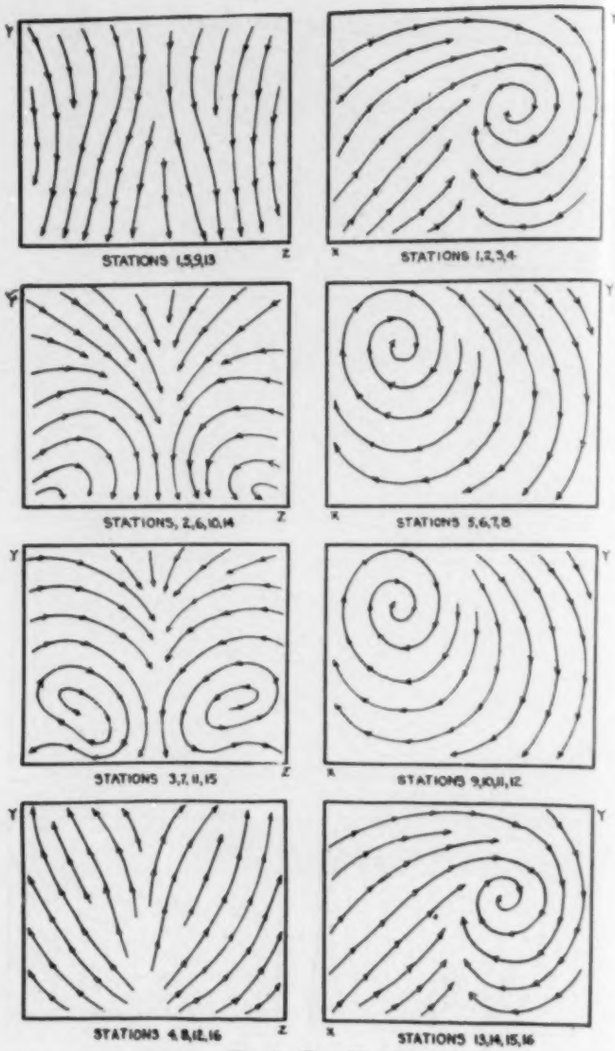


Fig. 8—Run No. 4

located symmetrically with respect to the wall of the y-z plane. The distribution plots for the planes of stations 1-2-3-4 and of stations 5-6-7-8 were symmetrical with those for the planes of stations 13-14-15-16 and of stations 9-10-11-12 respectively. The motion was principally defined by a horizontal swirl whose axis was approximately parallel to the z axis and which was located near the ceiling in the center of the room. The action of the air emitted from the inlet register was such as to concentrate the flow of air in its line of action. As the air struck the opposite wall it then went upward with a tendency to spread out again. Near the floor the motion was substantially parallel to the floor except in the far corners from the inlet duct.

Test Run No. 2. For the duct arrangement used for this test run the sweeping action of the air along the floor was not quite so pronounced as in the previous case. Directly in line with the inlet duct, of course, the air swept the floor in almost a straight line until it reached the opposite wall where it was deflected outward and upward. There was an upward motion of air along the wall opposite the wall of the ducts only over about three-quarters of its surface, as may be seen by glancing at the stream diagrams for the planes of stations 5-6-7-8, 9-10-11-12, and 13-14-15-16. The streamlines over the other quarter of that wall were inclined to be horizontal.

The significant factor of the air distribution for this duct arrangement as far as the occupants of the room would be concerned is the fact that in most of the zone of occupancy the air movement had a downward component, in many cases just as great as its horizontal component.

Test Run No. 3. Those planes on which velocity directions have been plotted for the center portion of the room indicate a very good mixing action in that vicinity. This turbulence would insure, in a full-sized installation such as this, the proper mixture of warm air entering the room from the register and cold air that might enter from any other source.

The vortex or swirl that was produced is about an axis that is approximately parallel to the wall of the inlet and outlet ducts (the y-z plane), but slopes upward from the wall of the x-y plane toward the opposite wall. There was a swirling, sweeping motion of the air along the floor with the tendency toward movement from the corner opposite the outlet duct toward that duct. Along that portion of the floor near the wall of the registers,

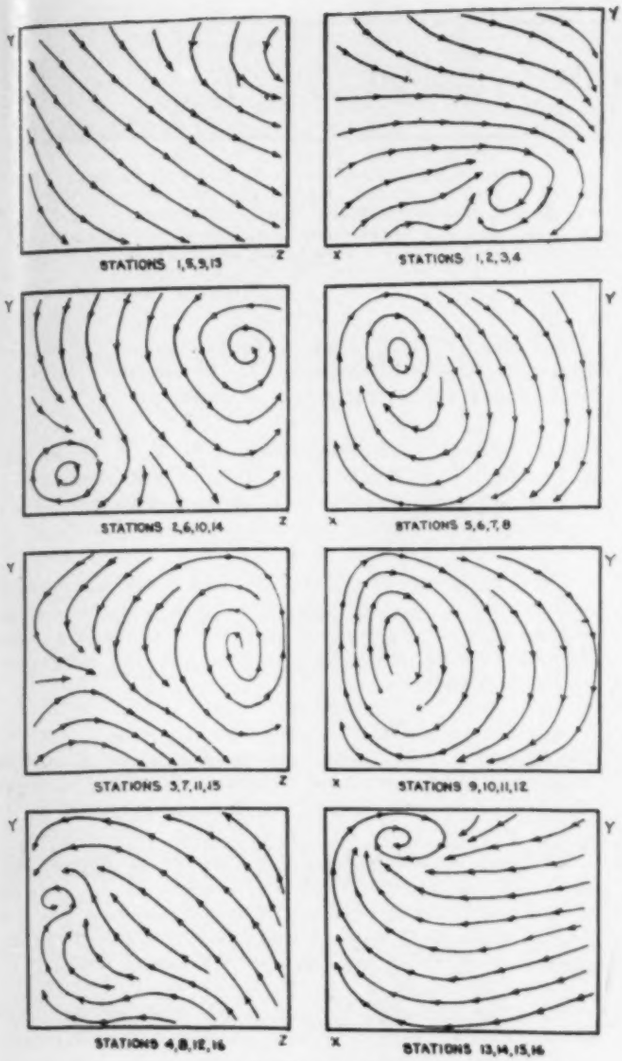


Fig. 9—Run No. 5

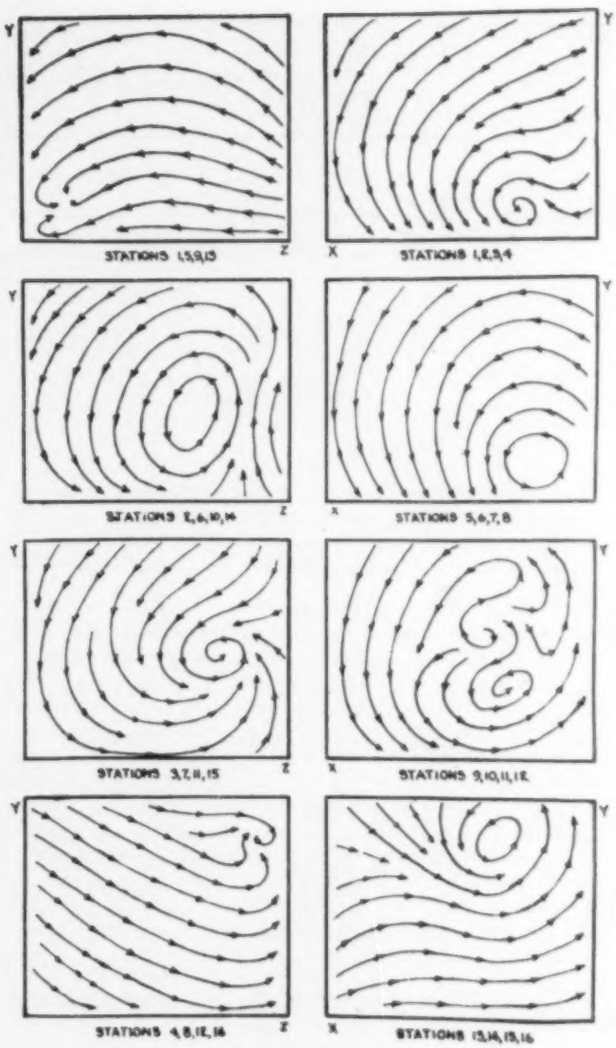


Fig. 10—Run No. 6

the air moved away from the outlet and began the formation of a swirl that extended up into the room.

Test Runs Nos. 4, 5, *and* 6. These test runs were made using the 1/5 scale model room, the duct arrangements being similar to those of the 1/10 scale model for Runs No. 1, 2, and 3 respectively. The models were geometrically similar in all respects and the same Reynolds' number was maintained in the entrance ducts in all cases. A comparison of the air distribution diagrams will show a striking resemblance between the stream patterns for the two models of different scale, but with similar duct arrangements. The fact that there is such good agreement in the flow patterns indicates the reliability of a scale model experiment in attempting to predict the behavior of air currents in a full-sized room.

Conclusions

The particular dimensions of the two scale models and the duct arrangements as used in this experimental investigation were not chosen because they necessarily typified common cases encountered in heating and ventilating installations. It was merely desired to investigate the effect a change in scale would have upon the air distribution within a room, and a simple case was chosen for analysis.

Considering the similarity of flow patterns obtained from the 1/10 and 1/5 scale model rooms, it would seem reasonable that flow similarity would also exist between a 1/10 or a 1/5 scale model room and the full-sized prototype. Then, if geometric similarity and Reynolds' number within the ducts were maintained in a scale model, the results obtained from it could justifiably be used in predicting the circulation of air within the prototype.

The placing of windows, the arrangement of the room for occupancy, and the comfort of the occupants of the room are all items that are more or less dependent on the manner in which the air circulates in the room. Simple scale model experiments would be a valuable aid in deciding upon the register arrangement to accomplish the desired circulation.

Acknowledgment

This article is a portion of a thesis completed in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering at the State University of Iowa at Iowa City. The experimental investigation and the writing of the thesis was carried out by the author under the direction of Prof. Huber O. Croft.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the A.S.H.V.E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A.S.H.V.E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

Controls in Residential Air Conditioning Work, by G. D. Kingsland. *Automatic Heat and Air Conditioning*, Vol. 9, No. 8, Aug. 1938, pp. 24-25 and 46-49. Description of simple, practical and comparatively inexpensive control systems for residential air conditioning installations.

•

How the Alternating Receiver Works on Vapor Systems, by Napier Adlam. *Heating and Ventilating*, Vol. 35, No. 8, Aug. 1938, pp. 19-23. Discussion of the principles of the alternating receiver on a vapor heating system and its proper location and function.

•

The Combustion Process in the Compression-Ignition Engine, by J. W. Drinkwater and Prof. A. C. Egerton. *The Institution of Mechanical Engineers, Proceedings*, London, Vol. 138, 1938, pp. 415-493. Report of tests carried on in the University Engineering Laboratory, Oxford, England to investigate the combustion process in a compression-ignition engine by determining the extent of the chemical reactions of the fuel and air at various stages during the compression and expansion strokes. Curves show the percentage volumes of the constituent gases in the engine cylinder at different points in the cycle. Attempts were made to determine the concentration of aldehydes in the gases, but the tests showed less than anticipated. Oxides of nitrogen were detected and were considered to have an influence upon cylinder wall corrosion.

•

The Internal Combustion Engine, by Dr. D. R. Pye. Vol. 1, Second edition, London, Oxford University Press. Recent results on fuels, dopes and detonations have been added to first edition.

•

Air Conditioning Fruit Cars with Diesels, by John R. West. *Ice and Refrigeration*, Vol. 95, No. 3, Sept. 1938, pp. 156-158. Precooling plays important part in shipment of perishable foods. Illustrated description of diesel precooling units. Reduction of fuel costs and savings in freight claims. Temperatures at various places in the cars and even in the centers of the crates are fairly uniform due to adequate air circulation.

•

The Temperature and Humidity Conditions of the Atmosphere in Mines at Great Depth, by J. H. Dobson and W. J. Walker. *The Colliery Guardian*, London, Vol. CLVII, Aug. 5, 1938, pp. 235-240. Discussion of the sources of heat underground and methods of air conditioning to insure health of workers and their ability to make a fair amount of physical effort. Ultra deep level mine air cooling is considered to be ahead of future requirements.

•

Mechanism of Heat Transfer by Evaporation, Allen G. Stimson. *Power Plant Engineering*, Vol. XLII, No. 8, p. 498, Aug. 1938, and ibid Vol. XLVII, No. 9, pp. 559-562, Sept. 1938. Discussion of the fundamental mechanism of the process of evaporation from the standpoint of the molecular theory. Surface tension and surface energy were studied and their effect on surface metability and vapor bubble origination. Observations and theory incident to the superheating of a boiling liquid, the critical sizes of bubbles and drops, the heat transfer coefficient and the tangible factors which influence the cofficient are stated.

Heat and Thermodynamics by Prof. Mark W. Zemansky. London, McGraw-Hill Publishing Co. The treatise is intended for students in engineering, physics or chemistry.

•

Silicosis and Similar Dust Diseases, by R. R. Sayers and R. R. Jones. *U. S. Treasury Dept. Public Health Reports*, Vol. 53, No. 33, Aug. 19, 1938, pp. 1453-1472. Summary of the knowledge concerning the cause and prevention of silicosis.

•

A Cooling Survey of Lincoln, Nebraska. *American Artisan*, Vol. 107, No. 8, Aug. 1938, pp. 37-55. Map of business section showing all air conditioning installations. Tonnage of air conditioning by field of application. Description of many commercial installations.

•

Attic Fan Installation Practices. *American Artisan*, Vol. 107, No. 8, Aug. 1938, pp. 61-64. 1938 practices covering attic fan installations to eliminate noise, vibration, whistle and to insure easy flow of air to the fan and out of the attic.

•

Windows. *Domestic Engineering*, Vol. 152, No. 3, Sept. 1938, pp. 48-49 and 110-112. Window conditioning is the term applied to the insulating techniques developed to prevent heat loss through windows. The use of double panes cuts down fuel costs, prevents condensation and eliminates unhealthful drafts.

•

Panel Heating Installation. *Automatic Heat and Air Conditioning*, Vol. 9, No. 9, Sept. 1938, pp. 30-32. Technical details on design and installation of a panel heating system in a 17-room residence.

•

House Construction and Insulation Affect Oil Burner Heating Loads, by Prof. J. D. Hoffman. *Air Conditioning and Oil Heat*, Vol. XI, No. 9, Sept. 1938, pp. 9-10 and 46. Discussion of insulation application to existing homes.

•

The Statistical Mechanics of Condensing Systems, by Max Born and Klaus Fuchs, Edinburgh. *Proceedings of The Royal Society, Series A, Mathematical and Physical Sciences*, No. 926, Vol. 166, June 3, 1938, pp. 391-414. Discussion of Mayers' mathematical explanation of the phenomena of condensation. It is a method of approximation starting from the gaseous state which leads to the discontinuity of the density on an isothermal curve which corresponds to condensation. The authors have made an effort to control and clarify these calculations and believe that they have shown rigorously and in a simpler way that Mayers' statements are completely correct.

•

Adsorption Dehumidification, Nature and Uses in Conditioning Air, by G. L. Simpson. *Refrigerating Engineering*, Vol. 36, No. 3, Sept. 1938, pp. 159-162. Solid adsorption involves the condensation of a gas on a solid. Discussion of adsorption of water vapor on solids which have two properties: (1) Air containing water vapor when brought into equilibrium with one of these adsorbents at atmospheric temperature is extremely dry; (2) The degree of dryness so obtained is but slightly affected by temperature throughout the usual atmospheric range. Also description of equipment for dehumidification of air.

NOMINATIONS FOR 1939

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1939, submits the following list of nominees:

For President:

J. F. McIntire, Detroit, Mich.

For First Vice-President:

F. E. Giesecke, College Station, Tex.

For Second Vice-President:

W. L. Fleisher, New York, N. Y.

For Treasurer:

M. F. Blankin, Philadelphia, Pa.

For Members of the Council:

Three-Year Term

A. J. Offner, New York, N. Y.
E. K. Campbell, Kansas City, Mo.
G. L. Tuve, Cleveland, Ohio.
S. H. Downs, Kalamazoo, Mich.

Respectfully submitted,

NOMINATING COMMITTEE,

H. H. Erickson, *Chairman*
C. Tasker, *Acting Secretary*

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. B-VIII—Section 11. The Nominating Committee shall consist of one (1) member eligible to vote designated by each Chapter, or his alternate also appointed by the Chapter. The Secretary of each Chapter shall certify to the Secretary of the Society on or before January first the names of the member and alternate selected.

The Committee shall meet at the Annual Meeting of the Society at the call of the Secretary of the Society and shall effect its own organization and elect its own Chairman. At the Semi-Annual Meeting of the Society, if possible, the Nominating Committee shall select the nominees for the ensuing year for the offices of President, First Vice-President, Second Vice-President, Treasurer, and four (4) members of the Council. In any event the names of the nominees shall be certified to the Secretary of the Society before September twentieth, with the written consent of each nominee to fill the office for which he has been selected and their names with the offices to which they have been nominated shall be published in the October issue of the Journal.

Art. B-IX—Section 2. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, eligible to vote, at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the Government of the Research Laboratory, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1939:

Three-Year Term

R. E. Daly, New York, N. Y.
W. A. Danielson, Louisville, Ky.
H. J. Rose, Pittsburgh, Pa.
C. Tasker, Toronto, Ont., Can.
C.-E. A. Winslow, New Haven, Conn.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

Article II—Organization

Section 1. *Committee on Research*—There shall be a standing committee known as the Committee on Research, consisting of fifteen (15) members each serving for three (3) years and five (5) retiring each year. The outgoing Chairman if serving in that capacity during the last year of his three (3) year term on the Committee shall without election become an additional member of the Committee on Research for one (1) year.

(*a*) The Council shall nominate previous to July first of each year five (5) members to fill the vacancies of those retiring at the next Annual Meeting.

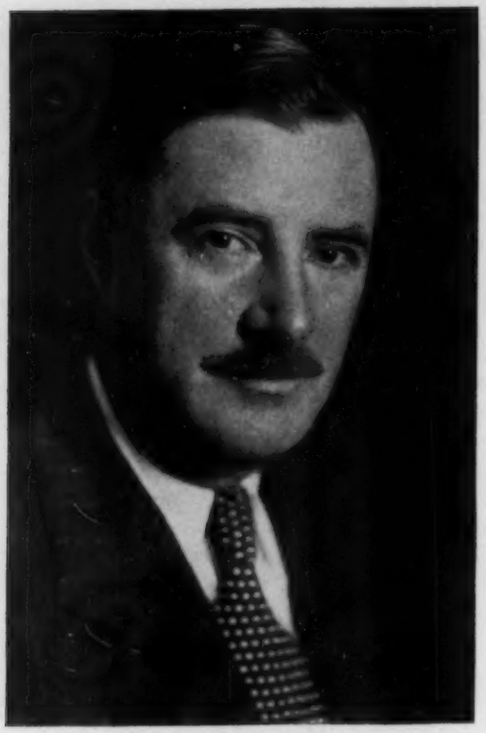
(*b*) The nominations made by the Council shall be published in the October issue of the Society's Journal.

(*c*) Any ten (10) members of the Society eligible to vote may present to the Secretary over their signatures, the name of one (1) or more additional nominees for the Committee on Research, provided such name or names are presented at least sixty (60) days prior to the next Annual Meeting, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

(*d*) The election shall otherwise conform to the regulations provided for the election of officers of the Society in the Constitution, By-laws and Rules.

(*e*) Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Nominees for A. S. H. V. E. Officers in 1939



J. F. McINTIRE
Detroit, Mich.
President

F. E. GIESECKE
College Station, Tex.
First Vice-President

W. L. FLEISHER
New York, N. Y.
Second Vice-President

M. F. BLANKIN
Philadelphia, Pa.
Treasurer

Council Members Nominated for Three-Year Term

E. K. Campbell
Kansas City, Mo.

S. H. Downs
Kalamazoo, Mich.

Alfred J. Offner
New York, N. Y.

G. L. Tuve
Cleveland, Ohio

A.S.H.V.E. Officers Address Chapters

Pres. E. Holt Gurney, of Toronto, inaugurated his fall visits to Society Chapters in October and the subject of his address was Air Conditioning Brings Human Problems to the Engineer. He visited the Montreal Chapter Oct. 6, and he will speak at meetings of the Washington, D. C. Chapter October 12, Philadelphia Chapter on October 13, and the New York Chapter on October 17. Mr. Gurney's speaking trip coincides with the opening of the 1938-39 Speakers Bureau.

J. F. McIntire, Detroit, First Vice-President of the Society, spoke on The Engineer at the Ontario Chapter meeting in the Royal York Hotel, Toronto, October 3, and he will address the Cincinnati Chapter meeting October 11. On October 6 Mr. McIntire and J. H. Walker addressed the Montreal Chapter.

The Speakers Bureau activity of the A. S. H. V. E. embraces talks on heating, ventilating and air conditioning by prominent authorities before the Chapters of the Society.

Prof. F. B. Rowley, past president of the Society and director of the Engineering Experiment Station of the University of Minnesota, will address the Michigan Chapter of the Society in Detroit October 10, and the Iowa-Nebraska Chapter on October 11 in Des Moines, on the subject of Condensation Within Walls.

Prof. C. P. Yaglou, assistant professor of industrial hygiene at the Harvard School of Public Health, will address the Michigan Chapter at Detroit November 7, and Northern Ohio Chapter, Cleveland, on November 8, and his subject will be The Relationship of Air Conditioning to Health.

W. A. Russell, Chairman of the Speakers Bureau Committee, announces that the following men have signified their willingness to serve on the Speakers Bureau:

H. E. Adams, South Norwalk, Conn.; Dr. A. A. Adler, Arlington, N. J.; Prof. A. B. Algren, Minneapolis, Minn.; Prof. E. A. Allcut, Toronto, Ont.; H. L. Alt, Long Island, N. Y.; C. T. Baker, Atlanta, Ga.; F. R. Bishop, Detroit, Mich.; Lincoln Bouillon, Seattle, Wash.; Prof. A. I. Brown, Columbus, Ohio; Albert Buenger, Dayton, Ohio; and Prof. E. F. Dawson, Norman, Okla.

J. A. Donnelly, Largent, W. Va.; Prof. Philip Drinker, Boston, Mass.; W. L. Dudley, Seattle, Wash.; Prof. M. K. Fahnestock, Urbana, Ill.; Prof. F. L. Fairbanks, Ithaca, N. Y.; Dr. M. B. Ferderber, Pittsburgh, Pa.; W. L. Fleisher, New York, N. Y.; R. E. Hattis, Chicago, Ill.; Dr. E. Vernon Hill, Chicago, Ill.; F. C. Houghten, Pittsburgh, Pa.; H. F. Hutzel, Detroit, Mich.; A. M. Kinney, Cincinnati, Ohio; Prof. A. P. Kratz, Urbana, Ill.; Prof. G. L. Larson, Madison, Wis.; and S. R. Lewis, Chicago, Ill.

E. C. Lloyd, Lancaster, Pa.; R. A. Miller, Pittsburgh, Pa.; Dr. C. A. Mills, Cincinnati, Ohio; Prof. D. W. Nelson, Madison, Wis.; Prof. B. F. Raber, Berkeley, Calif.; C. H Randolph, Milwaukee, Wis.; S. C. Rothmann, Charleston, W. Va.; Prof. F. B. Rowley, Minneapolis, Minn.; L. P. Saunders, Lockport, N. Y.; and Prof. R. R. Seeber, Houghton, Mich.

R. A. Sherman, Columbus, Ohio; A. E. Stacey, Jr., New York City; W. E. Stark, Cleveland, Ohio; Prof. G. L. Tuve, Cleveland, Ohio; T. H. Urdahl, Washington, D. C.; J. H. Van Alsburg, Chicago, Ill.; J. H. Walker, Detroit, Mich.; G. Lorne Wiggs, Montreal, Canada; Prof. G. B. Wilkes, Cambridge, Mass.; Prof. B. M. Woods, Berkeley, Calif.; and Prof. C P. Yaglou, Boston, Mass.

W. A. Oates Dies of War Injuries

The death of Walter A. Oates, lieutenant, U. S. Army retired, famous war aviator, and member of the American Society of Heating and Ventilating Engineers, occurred on July 30, 1938, at Lynn Hospital, Lynn, Mass., after a long illness.

Mr. Oates was born at Providence, R. I., on June 24, 1888 and studied at Columbia University and Massachusetts Institute of Technology. In November 1919 he became associated with Grinnell Co., Inc., with whom he was engaged in estimating, designing, drafting and as field engineer in charge of construction until May 1926. At that time he joined the staff of the Lynn Gas & Electric Co. in charge of the house heating department and he retained his affiliation with this company until his death. Despite his physical handicaps, the retired Army officer was one of the most popular employees during the years he had been associated with that firm and he was a familiar figure to thousands in Lynn and Swampscott. Mr. Oates became a Member of the A.S.H.V.E. in 1931 and he attended the 1937 Semi-Annual Meeting of the Society held at Swampscott, Mass. where he resided.

W. A. Oates

Mr. Oates joined the Aviation Corps as soon as war was declared by the United States, enlisting from New York City where he was employed as a meteorologist in the U. S. Weather Bureau. He was one of a group of 100 Americans chosen to be trained by the Royal Air Force for the purpose of taking command of American Air squadrons as they were formed. Lt. Oates was on detached service with the 107th Squadron, Royal Air Force, serving as flight commander of that squadron for several months. He was recognized as having one of the most outstanding records in the Army air service. His bravery and prowess in battle won him the Belgium Order of Leopold for the rescue of a Belgian brought down inside enemy lines; the French Medaille Militaire for bravery in action in escort work for French squadrons; and a recommendation for the English Distinguished Flying Cross, a decoration received by only seven Americans, in recognition of more than 1000 flying hours over the lines. Later he was awarded the Order of the Purple Heart by the United States government.

Lt. Oates participated in the Somme offensive in August of 1918, the engagement at St. Mihiel and the Meuse-Argonne. The engagement which resulted in the injuries that ultimately caused his death occurred while he was flying over the lines east of Cambria, France. He brought down three German planes before he was finally shot down himself, falling 9000 ft. He had a record of 12 planes and eight balloons, and served in three Royal Air Force squadrons on bombing, reconnaissance and pursuit work. His first severe wounds were received when he crashed after bringing down three of five enemy planes. In the fighting the controls on the right side of his plane were shot away and the gas tank was pierced. He landed out of control between the first and second lines. He was also shot down in flames twice during service. Because of his record he was given the right to fly in pursuit of enemy planes without specific orders.

He was a member of the Overseas Masonic Lodge of Providence, R. I.; East Lynn Post 291, American Legion; the One Hundred Aviators Club of New York; the Archie Club of World War Aviators of Boston; and the Disabled Emergency Officers of the World War.

Lt. Oates leaves his widow, Mrs. Dorothy Attwill Oates; a son, Joseph Attwill Oates, 15; his mother, Mrs. Elizabeth Oates of Providence; and a brother, Joseph H. Oates of Greenwood, R. I., to whom the Officers and Council have extended their sympathy.

Military honors were accorded Lieutenant Oates at funeral services held in Swampscott, where military leaders joined with business associates, friends and relatives in an impressive tribute to the flier. Among those who attended were representatives of the U. S. Army, and a representative of the British consulate at Boston. Three planes from the 101st Squadron, Massachusetts National Guard, flew overhead as the cortege made its way to Pine Grove Cemetery and during the services at the grave. A volley was fired by the East Lynn Post Drill Squad and taps were sounded by the post bugler.

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the American Society of Heating and Ventilating Engineers is located

45th Annual Meeting Marks 20 Years' Research

THE Society's 45th Annual Meeting marks the 20th anniversary of the establishment of its Research Laboratory in Pittsburgh. The meeting will be held January 23-26, 1939, in the William Penn Hotel and arrangements are in charge of R. A. Miller, general chairman.

A technical program is being developed that will strongly emphasize the research work being carried on by the Society and several papers will discuss the health aspects of air conditioning, air pollution, fuels and other allied subjects.

The Pittsburgh chapter members are planning an entertainment program that will be unique.

There will be an opportunity for the delegates from chapters to discuss problems of interest. Technical committees will have a day for meetings.

Reports of technical activities and Society business will be given at the sessions by the committee chairmen and the officers.

Pres. E. Holt Gurney will call the meeting to order on January 23 and this will focus attention on the work of the Society. He will review the highlights in its progress, paying fitting tribute to the founders of the laboratory.

Society members are justly proud of the fact that the A.S.H.V.E. is the only professional engineering organization that maintains and operates its own research laboratory. Every member has a direct interest in the work through his contribution from dues. The Committee on Research elected by the membership directs the work at the laboratory and in cooperating institutions and in the 20 years since the laboratory was established it has contributed much fundamental data to the profession and has been a primary factor in the advancement of the art and the industry.

At present there are 20 projects in research being carried on at the laboratory in the Bureau of Mines, Pittsburgh, and in cooperating institutions. Reports on the progress of this work will be given by Chairman W. L. Fleisher of the Committee on Research.

Because of the central location of Pittsburgh it is easily accessible to a large proportion of the Society's membership and the Pittsburgh chapter extends a most cordial invitation through Pres. John Collins, Jr. to all members to be in Pittsburgh for the 45th Annual Meeting January 23-26, 1939.

John Collins, Jr.

R. A. Miller

T. F. Rockwell

ANNUAL REPORTS OF LOCAL CHAPTERS

Illinois

A complete report of the Illinois Chapter activities has been prepared by Secy. M. W. Bishop.

The following officers were elected for the 1938-39 season and committees have been appointed by President Vernon:

President—J. R. Vernon.
Vice-President—Tom Brown.
Secretary—M. W. Bishop.
Treasurer—E. M. Mittendorff.
Board of Governors—I. E. Brooke, V. L. Sherman, J. H. Van Alsburg, and S. I. Rottmayer, past president.
Legislative Committee—John Howatt, Chairman; J. H. Milliken and J. J. Hayes.
Membership Committee—I. E. Brooke, Chairman; O. J. Greenwood, A. O. May, H. J. Prebensen and W. A. Stahl.
Meetings and Publications Committee—V. L. Sherman, Chairman; Tom Brown, R. E. Hattis, C. M. Burnam, Jr., and R. T. Miller.
Finance Committee—E. M. Mittendorff, Chairman; E. N. McDonnell, H. C. Mueller, F. I. Raymond, and H. C. Shilling.

The annual report of the Board of Governors of Illinois Chapter covering activities for 1937-38 is submitted and the following officers and members of the Board of Governors were elected at the meeting on May 10, 1937:

President—L. S. Ries.
Vice-President—S. I. Rottmayer.
Secretary—C. E. Price.
Treasurer—J. R. Vernon.
Board of Governors—M. W. Bishop, Tom Brown, E. M. Mittendorff, J. J. Hayes.

At the first meeting of the season Mr. Ries resigned as president due to his transfer from Chicago and this automatically moved Vice-President Rottmayer into the presidency. To fill the vacant office of vice-president, J. R. Vernon was elected. Tom Brown was then elected treasurer and O. W. Armspach was elected to the Board of Governors. The active officials for the majority of the year, therefore, were:

President—S. I. Rottmayer.
Vice-President—J. R. Vernon.
Secretary—C. E. Price.
Treasurer—Tom Brown.
Board of Governors—O. W. Armspach, M. W. Bishop, J. J. Hayes, E. M. Mittendorff.

The following committees were appointed to carry on the regular business of the Chapter and functioned throughout the year:

Membership—E. M. Mittendorff, Chairman; C. M. Baumgardner, B. A. Broom, J. T. Machen, G. G. Turner.
Finance—J. R. Vernon, Chairman; J. C. Matchett, E. W. Rietz, O. W. Armspach, M. W. Bishop.
Meetings and Publications—Tom Brown, Chairman; S. I. Rottmayer, H. P. Reid, V. L. Sherman, N. W. Swanson.
Legislative—R. E. Hattis, Chairman; John Howatt, J. H. Milliken.

In addition to the regular committees who served during the previous season, a special committee to represent the Chapter in a proposed organization of an Illinois Engineering Council, was appointed, consisting of R. E. Hattis, Chairman; I. E. Brooke, and J. R. Vernon.

The Membership Committee should be credited with a job well done in securing 28 new Chapter members in the year. The Chapter membership as of May 1938 was 159. The Board of Governors records with sorrow the deaths of R. B. Hayward, H. G. Kreissl and George Blanding.

The meetings held during the year, with the speakers, subjects of their talks and attendance, were as follows, with an average attendance of 114:

October 11, 1937. Subject—Influence of Air Conditioning Upon Public Health. Speaker—Dr. Lloyd Arnold, professor of bacteriology and public health, University of Illinois College of Medicine. Attendance—106.

October 21, 1937. A special joint meeting with the *A. I. E. E.* and *Western Society of Engineers*. Subject—Invading the Kingdom of Dust. Speaker—H. W. Penney, engineering manager, Westinghouse Electric and Manufacturing Co. Attendance—21 Chapter members.

November 8, 1937. Subject—Mechanical Absorbents and Their Uses in Air Conditioning. Speaker—F. R. Bichowsky, consulting engineer, Dow Chemical Co. Attendance—75.

December 13, 1937. Subject—Automatic Control of Refrigeration. Speaker—A. B. Schellenberg, Alco Valve Co., Inc. Attendance—80.

January 10, 1938. Subject—Panel Heating. Speaker—T. N. Adlam, chief engineer, Sarco Co., Inc. Attendance—135.

February 14, 1938. Subject—Fire Hazards in Connection with Heating, Ventilating and Air Conditioning. Speakers—J. T. W. Babcock, engineer, *Western Factory Insurance Association*, and R. E. Maginnis, special agent, American District Telegraph Co. Attendance—82.

March 14, 1938. Subject—Mechanical Hot Water Heating Systems. Speaker—Prof. G. L. Larson, University of Wisconsin, past president A. S. H. V. E. and representative at this meeting of the Society's Speakers Bureau. Attendance—135.

April 11, 1938. Subject—Trends in Forced Air Heating. Speaker—S. Konzo, special research professor, University of Illinois. Attendance—135.

May 9, 1938. Illinois Chapter closed its 1937-1938 season with a dinner-dance business meeting at the Bismarck Hotel. National President E. Holt Gurney and Mrs. Gurney were guests of honor at this meeting, and Mr. Gurney completely charmed his entire audience with a very fine talk on The Importance of the Society, Its Functions and Its Objectives. Also present were four past presidents of the A. S. H. V. E. from Chicago—S. R. Lewis, H. M. Hart, Dr. E. V. Hill, and John Howatt, who were introduced by Illinois Chapter President Rottmayer. One hundred and fifteen members, ladies and guests were present.

The meetings were held at the Brevoort Hotel throughout the season. Thanks are due W. H. Howe for supplying and operating the projection machine and the entire Chapter should know and appreciate that the fine newspaper publicity given to activities was due to V. L. Sherman's work.

The Chapter sent its secretary to the Semi-Annual Meeting of the Society at Swampscott, Mass., in June and to the Annual Meeting in New York in January. At both meetings the Illinois Chapter was well represented by a large number of its members and their wives. At the Annual Meeting J. R. Vernon and Tom Brown served as member and alternate respectively from the Illinois Chapter on the Society's Nominating Committee. S. I. Rottmayer, as president, was the Chapter's official delegate to this meeting.

Throughout the year the Board worked together and with the members in harmonious fashion and wishes in this report to express its appreciation for the whole-hearted support received from every member.

Kansas City

A report of Kansas City Chapter activities prepared by Gustav Nottberg indicates that there are now 73 members on the Chapter's roll.

New Officers have been elected for 1938-39 as follows:

President—A. L. Maillard.
Vice-President—J. E. Matthews.
Secretary—C. A. Flarsheim.
Treasurer—K. M. Stevens.
Board of Governors—Prof. A. H. Sluss, Gustav Nottberg, and H. E. Gould.

Nine meetings were held from September to May and a brief summary is given.

September 20, 1937. Subject—Professional Development of

the Engineer. Speaker—I. C. Crawford, dean of School of Engineering, University of Kansas. Attendance—22.

October 4, 1937. Subject—Air Conditioning a Windowless Building. Speaker—R. E. Hattis, consulting engineer, Chicago. Attendance—23.

November 8, 1937. Subject—Treatment of Coal. Speaker—Art Stock, Sinclair Coal Co. Attendance—27.

December 11, 1937. Joint Meeting with *A. S. M. E.* and *Operative Millers Association*. Subjects and Speakers—Research in Air Conditioning for the Milling Industry, at Kansas State College, by Dr. C. O. Swanson, Kansas State College; Air Conditioning Problems in the Milling Industry, by Edgar Miller, technical editor, *Northwestern Miller*. Attendance—20.

January 3, 1938. Subject—Physiologic and Health Factors as Related to Air Conditioning. Speaker—Dr. C. A. Mills, University of Cincinnati. Attendance—42.

January 26, 1938. Subject—Air Conditioning Problems and Their Solutions. Speaker—C. S. Leopold, consulting engineer, Philadelphia. Attendance—59.

March 7, 1938. Subject—Forecasting Weather for Kansas City. Speaker—E. J. Christie, hydrologic supervisor for the Missouri River Region. Attendance—22.

April 4, 1938. Subject—Intermittent Heating of Buildings. Speaker—Prof. E. F. Dawson, University of Oklahoma. Attendance—40.

April 28, 1938. Subject—Influence of Building Construction on Air Conditioning. Speaker—Prof. F. E. Giesecke, Agricultural and Mechanical College of Texas. Attendance—33.

The activities of the chapter were directed during the season of 1937-38 by the following officers and committees:

President—Prof. A. H. Sluss.
Vice-President—A. L. Maillard.
Secretary—Gustav Nottberg.
Treasurer—J. E. Matthews.
Board of Governors—L. R. Chase, W. L. Cassell, M. A. Disney.
Membership Committee—Emil Haas, Jr., Chairman; Carl Clegg, Gustav Nottberg.
Auditing Committee—M. A. Disney, T. H. Anspacher.
Nominating Committee—C. A. Weiss, Chairman; L. A. Stephenson, L. R. Chase.
Spring Banquet Committee—L. A. Stephenson, Chairman; K. M. Stevens, Henry Nottberg, Jr.

Massachusetts

Secy. H. C. Moore has prepared a summary of meetings held by the Massachusetts Chapter during the past season and reports that the following officers have been elected to serve for the coming year:

President—James Holt
Vice-President—R. M. Nee
Secretary-Treasurer—H. C. Moore
Board of Governors—D. S. Boyden, Philip Drinker, H. W. Frederick, Alfred Kellogg, L. J. McMurrer, J. F. Tuttle and the Officers

October 26, 1937. Meeting at Harvard School of Public Health. Speakers—Prof. Philip Drinker on Air Pollution in Air Conditioning; Manfred Bowditch of the Massachusetts Department of Occupational Hygiene. Attendance—34.

November 17, 1937. Meeting at Massachusetts Institute of Technology. Speaker—R. G. Stone of the Harvard Blue Hill Observatory on Weather Forecasting. Attendance—48.

December 14, 1937. Meeting at Massachusetts Institute of Technology. President's Night, Dinner Meeting. Speaker—D. S. Boyden, president of the A.S.H.V.E. on Economic Aspects of District Heating and a general report of Society activities. Attendance—40.

January 18, 1938. Meeting at Massachusetts Institute of Technology. Speakers—A. W. Moulder on Hot Water Heating Systems; H. L. Alt on Specifications and Contracts for Heating and Air Conditioning. Attendance—65.

February 15, 1938. Meeting at Massachusetts Institute of Technology. Subject—Temperature and Humidity Controls. Speakers—J. H. Barrett, T. F. McCoy and J. H. Colby. Attendance—80.

March 22, 1938. Meeting at Massachusetts Institute of Technology. Speakers—W. E. R. Karsten on Compression Refrigerating Systems; E. L. Champion on Water Vapor Systems of Refrigeration. Attendance—60.

April 12, 1938. Meeting at Massachusetts Institute of Technology. Speaker—C. W. Walton, Jr. on Air Conditioning at Rockefeller Center. Also a movie on Heat and Its Control. Attendance—60.

May 17, 1938. Meeting at Massachusetts Institute of Technology. Speakers—T. S. Hart on Air Distribution Through Grilles; H. F. Hagen on Fan Characteristics and Duct Work. Attendance—50.

Michigan

The Michigan Chapter of the A. S. H. V. E. enjoyed a very successful 1937-38 season under the chairmanship of Frank Feely. The total membership at the end of the year was approximately 154 members, associates and juniors.

At the May meeting of the Michigan Chapter, officers were elected to serve for the season of 1938-39 as follows, and chairmen of a number of committees have been appointed to take charge of activities:

President—F. J. Linsenmeyer
Vice-President—W. C. Randall
Secretary—G. H. Tuttle
Treasurer—J. S. Kilner
Board of Governors—F. R. Bishop, F. J. Feely, W. H. Old, and S. S. Sanford.

The following men will serve as committee chairmen:

Program—W. C. Randall
Membership—E. H. Clark
Reception—J. P. Schechter
Publicity—H. F. Hutzel
Affiliate Council of the Engineering Society of Detroit—F. J. Feely and G. D. Winans

A synopsis of the meetings held during the year is as follows:

October 11, 1937. The opening meeting of the 1937-38 season was held at the Wardell Hotel with approximately 75 members and guests present. J. F. McIntire, then second vice-president of the A. S. H. V. E., discussed Society Activities for 1938. Dr. C. A. Mills, professor of experimental medicine, College of Medicine, University of Cincinnati, discussed the possibilities of radiant heating and cooling in his talk on Physiologic and Health Factors as Related to Air Conditioning.

November 8, 1937. Prof. J. D. Hoffman, director of practical mechanics at Purdue University, was the guest speaker at the November meeting. Professor Hoffman summarized the progress which has been made in the warm air heating industry in a talk entitled Warm Air Heating.

December 13, 1937. Approximately 80 members and guests attended the regular December meeting at which the subject of Filters was discussed. J. H. Harpham, president, Independent Air Filter Co., discussed the Development and Uses of Various Types of Mechanical Air Filters; and G. L. Penney, research engineer, Westinghouse Electric & Manufacturing Co., discussed Electrostatic Cleaners.

January 17, 1938. The Michigan Chapter was the guest of the Detroit Edison Co. at the regular January meeting. There were approximately 125 members and guests present. Douglas Dow spoke on Some Observations of English Heating and Ventilating Practice. The feature movie Topper was shown.

February 12, 1938. The annual Valentine dinner-dance of the Chapter was held at Dearborn Inn. There were 152 members, guests, and friends present. Music was provided by Dave Diamond and his orchestra.

March 14, 1938. T. W. Weigele, house heating engineer of the Detroit City Gas Co., was the guest speaker for the March meeting. Mr. Weigele presented some of the problems involved in providing Detroit with natural gas.

April 28, 1938. The April meeting was National Society Night. E. H. Gurney, president of the Society, discussed Air Conditioning Brings Human Problems to the Engineer. J. H. Walker, Council member, and vice-chairman of the Research

Committee, reviewed some of the activities of the Research Committee and also the Research Laboratory at Pittsburgh.

May 26, 1938. The annual golf meeting and election of officers was held at Birmingham Golf and Country Club. There were 34 golf players present and 60 attended the dinner.

Minnesota

The Minnesota Chapter has a total membership of 105 and activities are in charge of the following Officers, Board of Governors and Committees for 1938-39:

President—J. E. Swenson
Vice-President—F. C. Winterer
Secretary-Treasurer—M. H. Bjerken
Board of Governors—R. E. Backstrom and F. H. Schernbeck
Program Committee—F. C. Winterer, Chairman; A. B. Algren, R. E. Backstrom, G. S. Bean, H. M. Betts, E. F. Jones, F. W. Legler, William McNamara, H. S. Morton, F. H. Schernbeck and M. S. Wunderlich
Nominating Committee—R. E. Backstrom, Chairman; N. D. Adams and C. E. Gausman
Publicity Committee—F. F. Lange, Chairman, and D. B. Anderson
Entertainment Committee—H. P. Wallace, Jr., Chairman; N. D. Adams, D. S. Knapp and W. F. Uhl
Legislative Committee—F. B. Rowley, Chairman; H. M. Betts and E. F. Jones.
Auditing Committee—C. E. Gausman, Chairman, and L. C. Gross
Membership Committee—D. M. Forfar, Chairman; G. A. Dahlstrom, C. D. Fitts, C. E. Lund and H. C. Oberg
Attendance Committee—D. W. Hickey, Chairman, and J. R. Hall
Advisory Committee—R. E. Backstrom and all Past Presidents

A synopsis of meetings held during 1937-38 is presented by Secy. M. H. Bjerken:

September 20, 1937. Inspection, Sanitary District Treatment Plant, C. C. Wilbur, chief engineer. Attendance—35.

October 11, 1937. Inspection trip, Minneapolis-Honeywell Regulator Co. Attendance—70.

November 8, 1937. History of the Art—Method and Equipment of Future Trends in Air Conditioning, E. L. Hogan. Attendance—65.

December 13, 1937. Mechanical Hot Water Heating, Prof. G. L. Larson. Attendance—65.

January 10, 1938. Mechanical Forced Air Heating, B. F. McLouth. Attendance—40.

February 14, 1938. Condensation in Walls, Prof. F. B. Rowley, University of Minnesota. Attendance—100.

March 14, 1938. Trend of Air Conditioning, J. F. Murphy, General Electric Co. Attendance—55.

April 11, 1938. Coils for Cooling—Dehumidification, William Goodman, Trane Co., La Crosse, Wis. Attendance—85.

May 16, 1938. Annual Meeting at Golden Valley Golf Club. During the afternoon bridge was played by the ladies and those men who did not play golf. Officers were elected and after the business meeting adjourned, members and guests assembled in the dining room for dinner and dancing and distribution of golf, bridge and attendance prizes. Attendance—130.

New York

New York Chapter has elected the following officers to serve during 1938-39:

President—H. G. Meinke
Vice-President—O. O. Oaks
Secretary—T. W. Reynolds
Treasurer—W. M. Heebner
Board of Governors—H. L. Baker, Jr., W. E. Heibel, and C. S. Pabst.

President Meinke has announced the appointment of the Membership Committee consisting of C. S. Pabst, Chairman; W. T. Wrightson, and H. Gitterman; and O. O. Oaks as Chairman of the Program Committee.

The Chapter has 172 members and held regular meetings during 1937-38 as follows:

October 18, 1937. Col. D. S. Boyden in his annual visit to the New York Chapter presented details of the district heating system of the Boston Edison Co. He told how the varied demands of customers are met in buildings of dissimilar nature and presented figures regarding savings effected by the reclamation of condensate.

November 15, 1937. Dr. T. R. Thorburn, assistant surgeon of the New York Opthalmic Hospital addressed the Chapter on the subject What Patients Tell Their Doctors About Air Conditioning.

December 20, 1937. This evening was taken up by the annual dinner and bridge party arranged by H. W. Fiedler and brought out an attendance of 170.

January 17, 1938. G. T. Stanton, manager, technical consulting dept., Electrical Research Products, Inc., New York, told how the many problems of sound and of noise prevention have been met by the organization with which he is connected.

February 21, 1938. The subject of this meeting was Application and Economy of Steam Jet Refrigeration to Air Conditioning. Speakers—A. R. Mumford, research engineer, and A. A. Markson, assistant research engineer, New York Steam Corp.

March 21, 1938. This was an interesting meeting and many members joined in the discussion as to how they would heat the small six-room dwelling house, plans of which had been previously mailed to all members.

April 18, 1938. A movie tone on the Romance of Iron and Steel showed the continuous process of rolling sheets of The American Rolling Mill Co.

May 16, 1938. The annual dinner and entertainment of the Chapter was held on this date. R. E. Vetterle, special agent in charge of Federal Bureau of Investigation, New York City, gave an interesting account of the many problems encountered in the every-day work of a special agent. The attendance was 245.

Ontario

Ontario Chapter looks forward to another successful year under the leadership of the following officers:

President—H. B. Jenney
Vice-President—H. D. Henion
Secretary-Treasurer—H. R. Roth
Board of Governors—F. E. Ellis, J. W. O'Neill, G. A. Playfair, and C. Tasker

Secy. H. R. Roth reports that there are 108 members on the Chapter roll and that committees for the coming year have been selected as follows:

Attendance Committee—J. H. Fox, Chairman; L. P. Baker, Herbert Barnes, Ernest Fox, H. G. Hill, G. H. Hopper, S. A. Jennings, J. J. Lawlor, R. S. Mathison, William Philip, C. D. Waldon
Papers Committee—C. Tasker, Chairman; W. R. Blackhall, A. J. Dickey, J. H. Fox
Auditors—A. H. Hills, D. J. MacDonald
Entertainment Committee—E. R. Gauley, Chairman; W. P. Boddington, G. E. Cole, A. G. Ritchie.
Greeters Committee—W. P. Boddington, Chairman; L. P. Baker, J. P. Fitzsimmons, L. K. Hughes.
Membership Committee—S. W. Alexander, Chairman; J. H. Daynes, W. D. Gordon, E. R. Gurney, R. S. Maxwell, H. S. Moore, A. S. Morgan, D. O. Price.

The Chapter meetings were devoted to the following subjects:

October 12, 1937. Speaker—Prof. C.-E. A. Winslow, Yale University, on The Influence of Air Conditions on Human Health and Comfort. Attendance—64.

November 16, 1937. Speaker—Col. D. S. Boyden on District Heating System of the Boston Edison Co. Attendance—55.

December 6, 1937. Speakers—S. L. Elmer, Jr. on Methods for Comfort Cooling in Ontario; and J. P. Fitzsimmons on The Use of Water Cooling Coils in Comfort Cooling. Attendance—81.

January 10, 1938. A joint meeting was held of the local chapters of the *American Society of Mechanical Engineers, the Engineering Institute of Canada* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Speaker—Prof. E. A. Allcut of the University of Toronto, on Heat Insulating Materials and Their Properties. Attendance—168.

February 7, 1938. Speaker—Prof. Philip Drinker of the Harvard School of Public Health on Air Pollution and Dust Control. Attendance—53.

March 7, 1938. Speaker—A. L. Waines on Heat. Attendance—67.

April 4, 1938. Speaker—D. J. Purinton on Radiant Heating. Attendance—68.

May 7, 1938. Annual Meeting was held. Attendance—43 members.

Western Michigan

Western Michigan Chapter held meetings regularly during the past year and reports a membership roll of 35.

New officers and committees are serving for the 1938-39 season as follows:

President—C. R. McConner
Vice-President—B. F. McLouth
Secretary—W. G. Schlichting
Treasurer—H. J. Young
Board of Governors—W. W. Bradfield, S. H. Downs and S. W. Todd, Jr.
Speakers and Educational Committee—S. H. Downs, Chairman; Prof. L. G. Miller and T. L. Osberger
Entertainment Committee—S. W. Todd, Jr., Chairman; C. H. Pesterfield, F. C. Warren, and R. W. Wilson
Publicity and Research Committee—H. A. Brinker, A. B. Epple, O. D. Marshall, and B. F. McLouth
Nominating Representative, A. S. H. V. E.—W. W. Bradfield, S. H. Downs, Alternate
Nominating Committee, Local Chapter—O. D. Marshall, Chairman; W. W. Bradfield, and Prof. L. G. Miller
Membership Committee—W. W. Bradfield, Chairman; Arthur Boot, J. S. Cunningham, H. J. Metzger, H. F. Reid, and H. S. Witmer
Heating, Ventilating and Air Conditioning Code Committee—Prof. L. G. Miller, Chairman; A. B. Epple, B. F. McLouth, W. G. Schlichting, and K. L. Zeisse

Secy. W. G. Schlichting has presented a synopsis of the meetings in his annual report:

October 11, 1937. Rowe Hotel, Grand Rapids. Speaker—R. L. Davis on Sound Measurements and Related Practical Problems. Attendance—67.

November 8, 1937. Mertens Hotel, Grand Rapids. Symposium on Heating. Speakers—V. C. Deane on Hot Water; S. L. Worthing on Steam; Prof. L. G. Miller on Forced Warm Air. Attendance—70.

December 13, 1937. Columbia Hotel, Kalamazoo. Speaker—H. E. Birkholz on Air Filtration in Theory and Practice. Attendance—69.

January 10, 1938. Mertens Hotel, Grand Rapids. Speaker—Dr. E. V. Hill on The Effect of Air Conditions on Comfort, Health and Disease. Attendance—88.

February 14, 1938. Mertens Hotel, Grand Rapids. B. F. McLouth, S. H. Downs, Prof. L. G. Miller, O. D. Marshall and President Bradfield reported on several papers presented at the Annual Meeting. Attendance—77.

March 30, 1938. Joint meeting with Michigan Chapter at the Forced Warm Air Short Course, Michigan State College, East Lansing. Speaker—J. F. McIntire on Research and Air Conditioning—Magic Words. Attendance—248.

April 25, 1938. Mertens Hotel, Grand Rapids. Symposium on Fuels. Speakers—Dean H. B. Dirks on Coal; C. H. Pesterfield on Oil; C. Robbins on Gas. Attendance—70.

May 19, 1938. Green Ridge Country Club, Grand Rapids. Annual Meeting and Election of Officers. Attendance—160.

Atlanta Chapter's Athletes Compete

Our special correspondent in Atlanta, Ann Onymous, has recorded the minutes of a special meeting held on July 16. The report says, "In the Year of Our Lord Nineteen Hundred and Thirty-eight, a gathering of the Clan of Elbow Benders from the A. S. H. V. E. Atlanta Chapter occurred near a small lake on the outskirts of the City of Atlanta. Inasmuch as only 28 out of an expected 30 guests made their appearance, it became necessary to levy a small assessment on those present. However, the Chapter's able secretary, H. King McCain, was equal to the occasion and overcame all objections to the slightly increased tariff through whispering, 'Government Tax.'

A candid camera study of Pres. Templin explaining to Past Pres. Klein why he prefers Coca Cola.

"Pastimes enjoyed were swimming and horseshoe pitching. However a game that found universal favor and produced much heat and humidity was the Tall Story Contest for which only competitors were eligible.

"The official photographer, Candid Camera, was there in the person of Vice-Pres. S. W. Boyd, who has submitted the accompanying proof of the serious theories propounded at this meeting. It is requested that it be noted from the expressions of Pres. C. L. Templin and Past Pres. E. W. Klein that, 'the look of innocence on their faces has went and in its place is sweet content.' The meeting adjourned when the supply of Coca Cola and other beverages gave out."

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 13 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council,